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Results of Heating Mode Performance Tests of a Solar-Assisted Heat Pump

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Results of Heating Mode Performance Tests of a Solar-Assisted Heat Pump

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Scientific and Technical
Information Office

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SUMMARY

The performance of a heat pump, utilizing 8.16 square meters of low-cost solar collectors as the evaporator in a Freon-114[1] refrigeration cycle, was determined under actual insolation conditions during the summer and fall of 1976. C.O.P.'s greater than 3 were obtained with condensing temperatures around 78°C and evaporating temperatures around 27°C. Ambient temperatures were about 3°C above evaporating temperatures. Similar performance levels were obtained at other insolation and temperature conditions. Experience with the system has identified some component and system changes which should increase the obtainable C.O.P. to about 4.0. These are described along with the system's design rationale. The accumulated data are presented as an appendix.

[1] Use of commercial products or names of manufacturers in this report does not constitute official endorsement of such products or manufacturers, either expressed or implied, by the National Aeronautics and Space Administration.

INTRODUCTION

Due to growing uncertainty regarding the availability of oil, natural gas, and propane, it seems probable that the reliance on electrical power for residential space heating will increase substantially in the future. To provide the anticipated capacity, electric utilities must initiate construction of additional power generation plants, thus tending to further increase a homeowner's bill already inflated by high fuel costs. In an effort to partially offset the effects of such growth, utilities and manufacturers are promoting the use of electrically-driven, vapor compression heat pumps. Typical air-to-air heat pumps, however, are generally suitable for heating only in areas with moderate winter temperatures, the power required for a given output being primarily a function of the temperature difference between the heat reservoir and the heat sink. In fact, as this temperature difference increases (ambient temperatures drop below about $0^{\circ}C$), the system's performance is reduced to the point that it is providing little more than resistance heating. Water and earth heat pumps have been developed but with limited acceptance and range of application. Consideration of these facts has prompted the concept of utilizing solar energy as a heat source for wintertime operation, thus increasing the geographical applicability of the heat pump as well as reducing the power requirement (especially if adequate thermal storage is provided). The solar collectors may also be employed as a heat sink for cooling mode operation by rejecting heat at night through convective and radiative heat exchange with the ambient air and the night sky.

A brief review of refrigeration cycles would perhaps be helpful. The theory of operation of the heat pump is essentially based on the reversed Rankine cycle as shown on the P-h and T-S diagrams given in Figures 1 and 2, respectively. Referring to the basic equipment layout shown in Figure 3, the refrigerant enters the compressor as a low pressure, saturated vapor at point 1. In the compressor it is isentropically compressed to the superheated condition at point 2. From point 2 to 3 the high pressure vapor is condensed, giving up the latent heat of vaporization to the heat sink. The high pressure, saturated liquid leaving the condenser enters an expansion valve at point 3 where the refrigerant expands to a lower pressure prior to entering the evaporator at point 4. In the evaporator, the wet mixture vaporizes completely, absorbing the latent heat of vaporization from the heat reservoir. The saturated vapor leaving the evaporator again enters the compressor at point 1 as the cycle continues.

The performance of a refrigeration machine is generally evaluated by means of the Coefficient of Performance (COP), defined as the ratio of the useful refrigerating effect to the power requirement of the compressor. For the heating mode operation of a heat pump, the useful refrigerating effect is the heat rejected in the condenser, whereas for cooling mode operation it is the heat absorbed in the evaporator. Refering to Figure 1, the COP's for the heating and cooling modes can be seen to be given as follows:

$$COP_h = \frac{h_2 - h_3}{h_2 - h_1} \quad : \quad COP_c = \frac{h_1 - h_4}{h_2 - h_1}$$

where h_i refers to the refrigerant enthalpy at point "i". The maximum value the COP may obtain for some operating condition is that of a

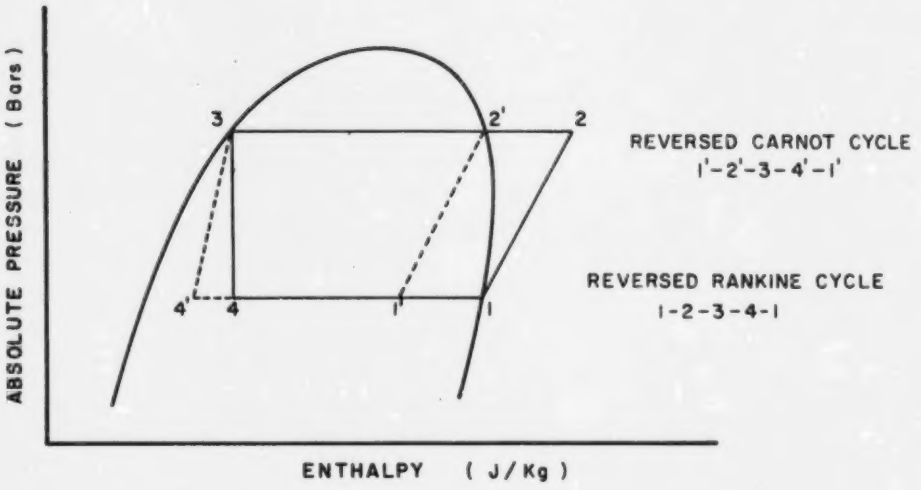

FIGURE 1. Pressure-Enthalpy Diagram. (1 bar = 10^5 Pa.)

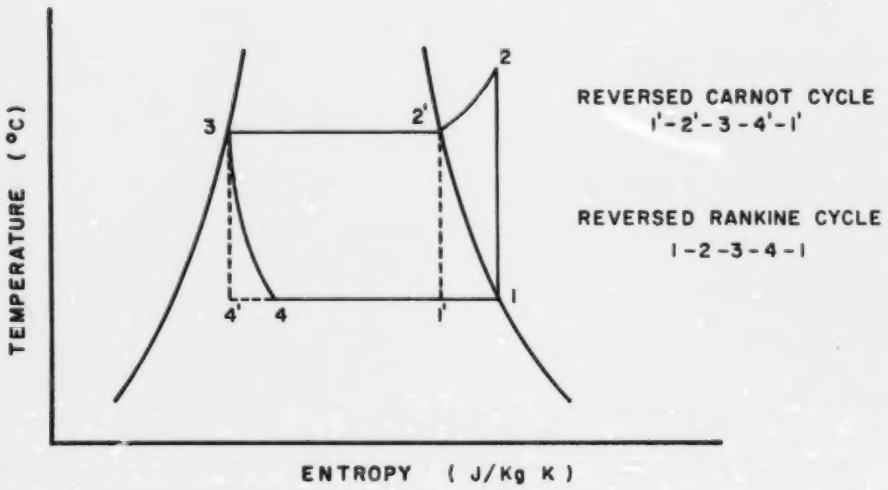

FIGURE 2. Temperature-Entropy Diagram.

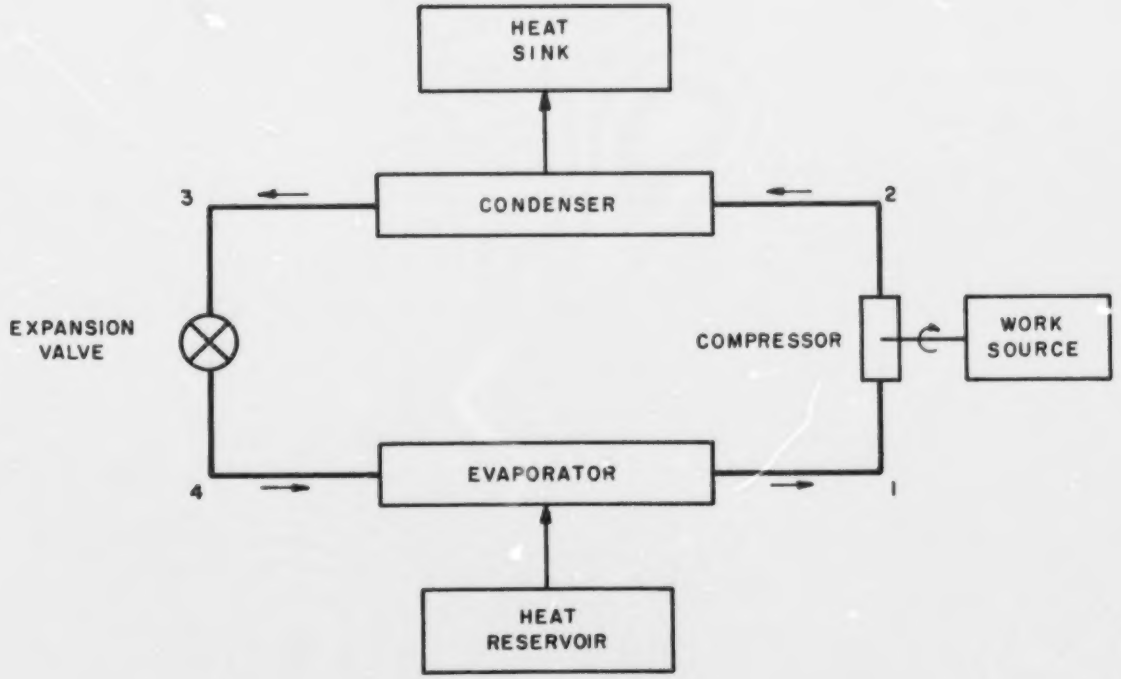

FIGURE 3. Basic Refrigeration Component Layout.

reversed Carnot cycle (dotted lines in Figures 1 and 2) operating at the same evaporating and condensing temperatures. The reversed Carnot cycle COP's are given as follows:

$$\text{REVERSED CARNOT COP}_h = \frac{T_3}{T_3 - T_1} \quad : \quad \text{REVERSED CARNOT COP}_c = \frac{T_1}{T_3 - T_1}$$

Thus, the smaller the difference between the evaporating and condensing temperatures, the greater the COP. This conclusion also holds true for the actual as well as the reversed Rankine refrigeration cycle.

Solar-assisted heat pumps present two basic design alternatives. Consider the heating mode operation. In one scheme, a conventional water (or air) solar collection system (with thermal storage) supplies energy to the heat pump evaporator (see Figure 4). Typically, this is done by circulating water (or air) in a closed loop from the collector to the thermal storage. The heat pump then extracts energy from the storage and delivers it to the load at the required temperature. The second design alternative utilizes solar energy directly, the solar collectors functioning as evaporators and with thermal storage being provided on the condenser side of the heat pump (see Figure 5). Generally, for cooling mode operation, the heating mode functions of the evaporator and condenser are reversed for both design options, the heat sink being provided by a cooling tower or forced air condenser. Systems based on the former concept (low side storage) have been developed by Yanagimachi in the Tokyo area and by Bliss in Tucson, Arizona. These systems both use water as the working fluid in unglazed, flat plate solar collectors. Alcone [1] presented an analytical investigation on the use of air-type solar collectors with rock-bin storage for use in conjunction with air-to-air heat pumps.

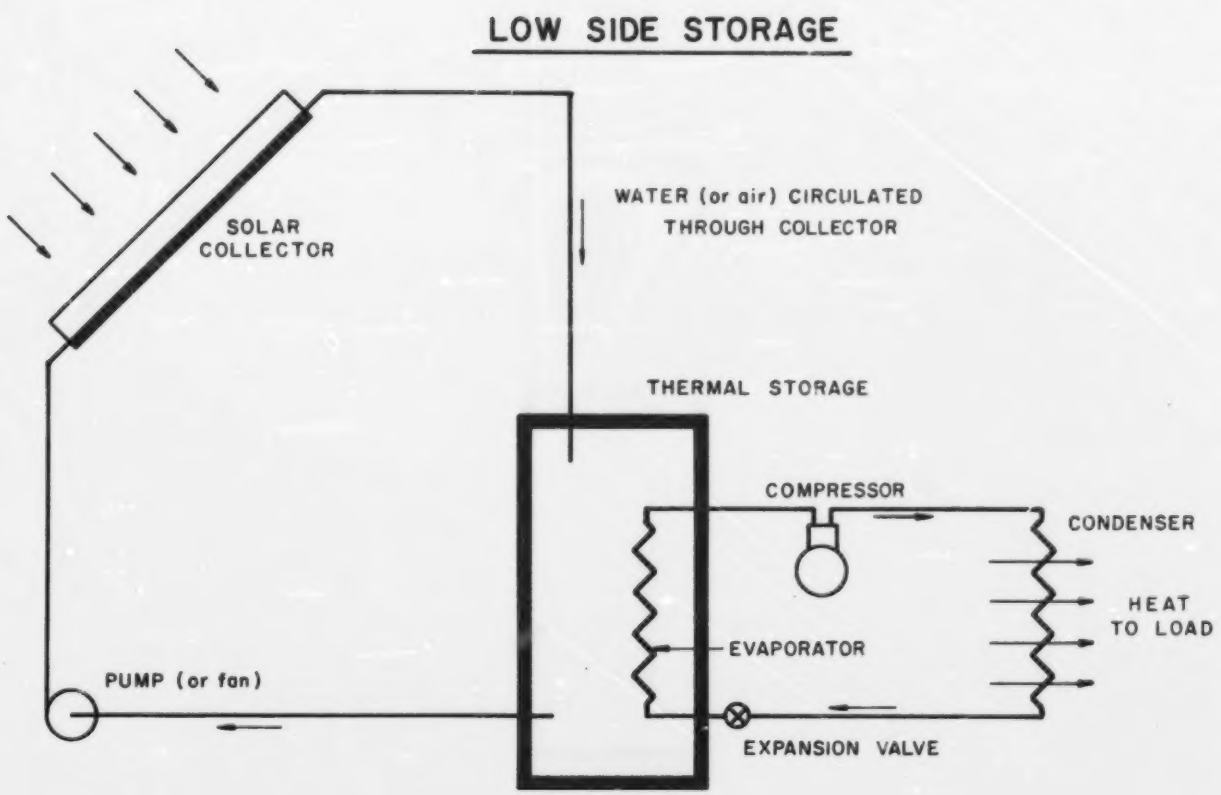

FIGURE 4. Low Side Storage Configuration of Solar-Assisted Heat Pump in Heating Mode.

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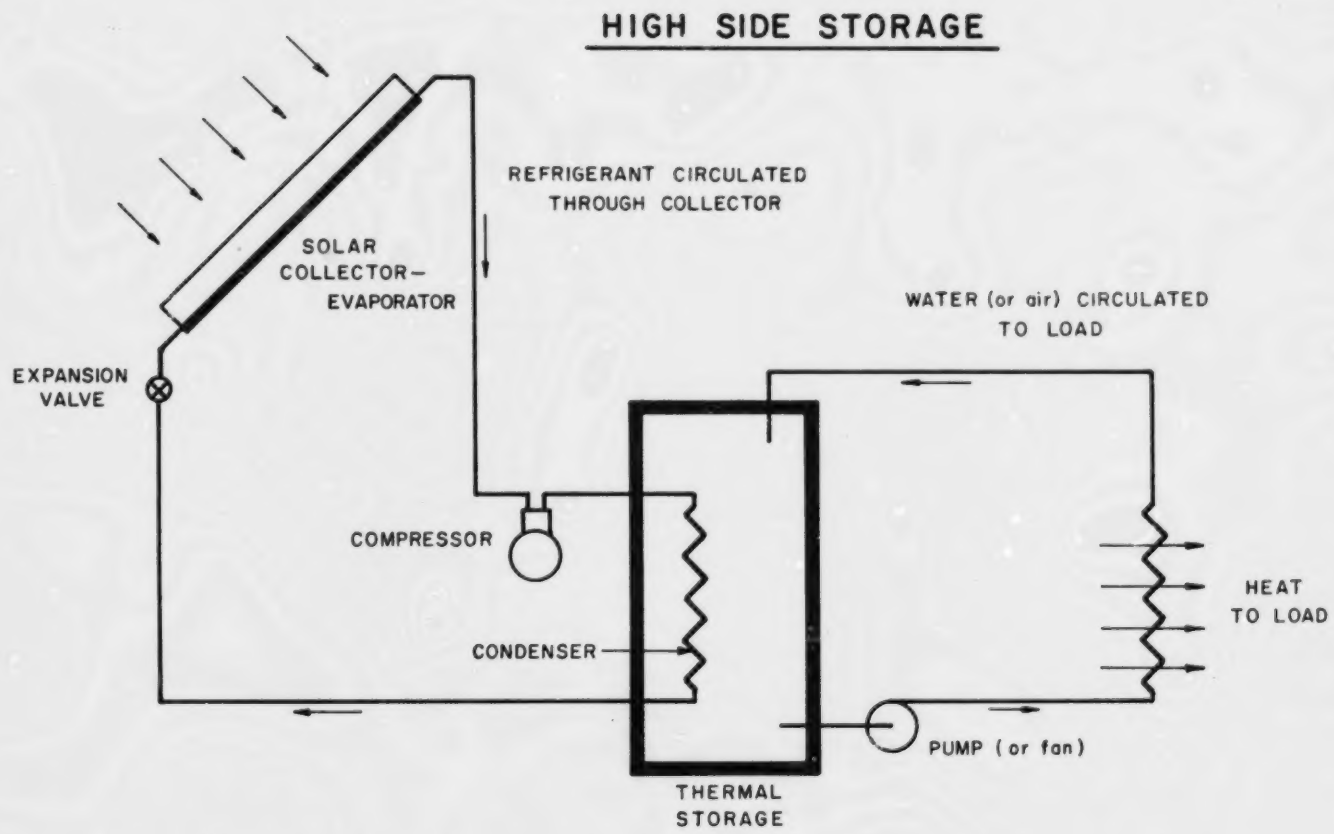


FIGURE 5. High Side Storage Configuration of Solar-Assisted Heat Pump in Heating Mode.

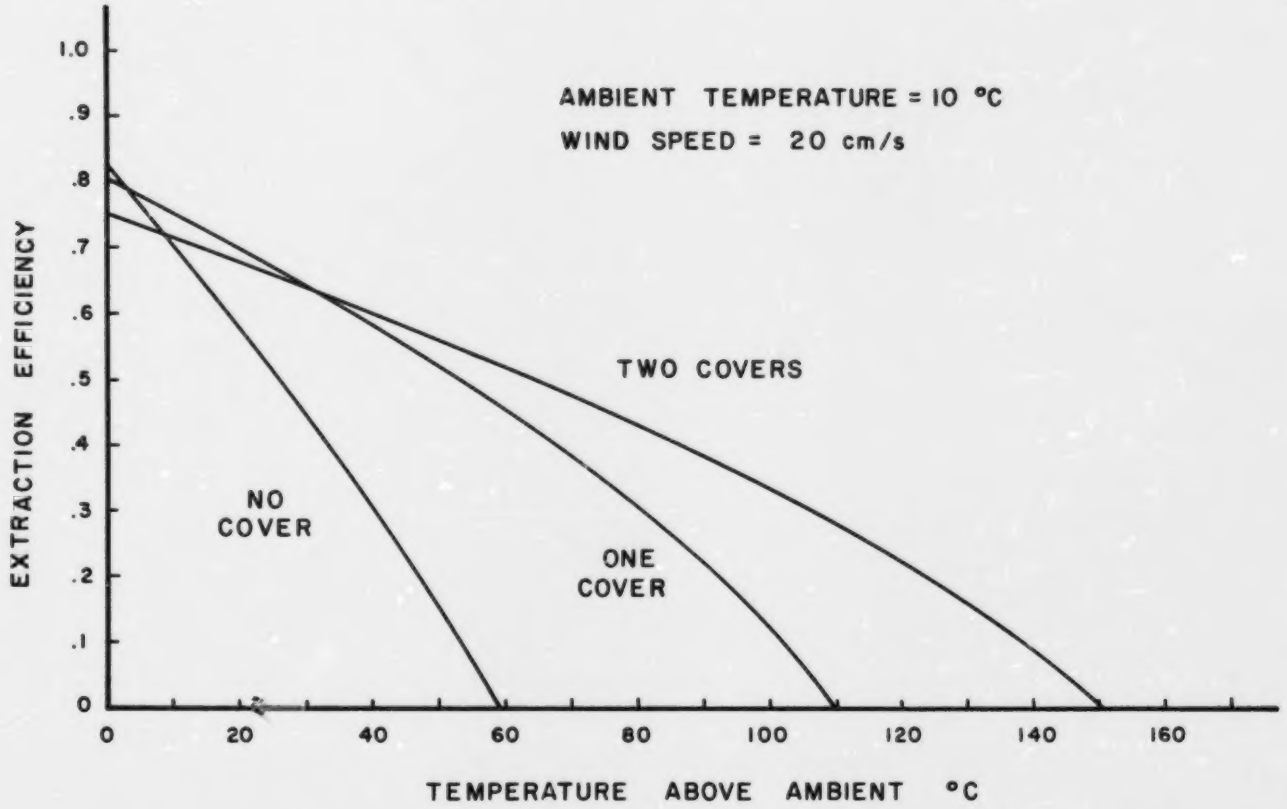
The heating mode of the second design scheme (high side storage) was explored by Sporn and Ambrose in a system constructed in New Haven, West Virginia. This alternative was also investigated in a system built by mechanical engineering students under the direction of Dr. F. O. Smetana at North Carolina State University at Raleigh. The results of heating mode performance testing of this system is the subject of this report.

The solar assisted heat pump system developed at North Carolina State was originally a subsystem of an alternative energy package designed and built by ten senior mechanical engineering students for participation in a national competition organized by SCORE, Inc. (Student Competitions on Relevent Engineering) and held in Albuquerque, New Mexico on August 12-16, 1975. After return from New Mexico, the system was partially modified for demonstration at the 1975 North Carolina State Fair under funding provided by the North Carolina General Assembly. The background and results obtained up to this time were reported in NASA CR-2771, "Solar Assisted Heat Pumps: A Possible Wave of the Future", December 1976. Subsequently, the heat pump portion of the system was instrumented and tested in detail. In this report, the various system modifications made during the course of performance testing will not be described in detail. Pertinent design considerations as reflected in the above mentioned changes, however, are discussed in a later section.

DESIGN RATIONALE

As pointed out in the previous section, the COP of a heat pump varies with the difference between the evaporating and condensing temperatures. Hence, for a given condensing temperature and output, the power requirement is reduced as the evaporating temperature is increased; the compressor does not have to pump the energy as far "uphill". The performance of a solar collector, on the other hand, is basically a function of the difference between the mean collector temperature and the ambient temperature. This relationship is shown graphically in Figure 6 for a flat plate collector with different numbers of cover plates. The extraction efficiency is defined as the ratio of useful energy removed from the collector to the solar energy flux normal to the collector surface. Thus, for a given ambient temperature, it is advantageous to operate the collector at as low a temperature as possible while still satisfying the desired output requirements.

Consideration of the relationships stated above suggests that the performance of both the solar collector and the heat pump can be materially improved if the two are combined in a single system. While an increase in overall system efficiency is the primary attraction of the solar assisted heat pump, several additional advantages may also be realized. First consider the solar collector. As operating temperatures are dropped, the collector becomes more efficient, thereby reducing the total area required for a given output. Additionally, one cover plate, or possibly no cover in milder climates, is generally sufficient to maintain acceptable performance. These factors can result in substantially lower collector weight and costs, especially in large installations.



NOTE: INFORMATION SHOWN IN THIS FIGURE TAKEN FROM APPLIED SOLAR ENERGY BY MEINEL AND MEINEL, pg. 433.

FIGURE 6. Flat Plate Solar Collector Performance.

Since a halocarbon type refrigerant is most likely to be used in the heat pump, the collector is free from freezing in cold climates, a serious problem in collectors using water as the working fluid. The use of a halocarbon refrigerant also permits the utilization of aluminum for the construction of the absorber panel without fear of corrosion or electrolysis (if dissimilar metals are used for piping connections). Prefabricated aluminum panels are readily available, light in weight, and relatively inexpensive, thus further contributing to the reduction of overall collector system cost. The heat pump also allows the collection of solar energy during unfavorable conditions (although at the expense of increased power consumption) which can permit a reduction of the thermal storage required.

With regard to the heat pump itself, the principal advantages of utilizing solar energy as the heat source have already been discussed, namely improved performance (reduced power requirements) and increased range of applicability. Another consideration is that lower temperature differences between the condenser and evaporator are accompanied by a reduction of the pressure ratios across the compressor. This should result in extended compressor life and increased reliability as there is less strain on the compressor components.

An additional possible advantage of a solar heat pump installation arises with regard to the daily distribution of load demands on the electric utility. The heat pump avoids typical residential demand peaks by operating during the day in winter and at night in the summer. Indications are that peak pricing policies may become a reality in the future. If so, this factor would serve to increase the attractiveness of the solar assisted heat pump.

The major drawbacks suffered with the use of the heat pump system are increased mechanical complexity, high initial cost, and higher operating costs. While in relatively large installations, the reduction in collector cost should more than offset the additional costs incurred with the heat pump system, smaller installations may not fare as well. The requirement for power to drive the compressor may not be acceptable for some applications, although this energy is nearly completely recoverable during heating mode operation. The increased mechanical complexity of the heat pump system not only influences the initial cost, but demands periodic maintenance as well. Another drawback is that operation of the system under the high pressures required with most refrigerants increases the probability for leakage, an important economic as well as environmental consideration.

In comparison with the relative simplicity of conventional water or air solar collection systems, the disadvantages stated above may appear serious. The situation takes on a different light, however, when consideration is given to the fact that both heating and cooling are provided by the heat pump system. A cost and performance comparison study by Rosar between a R-22 solar assisted heat pump and an ammonia absorption cooling and heating system concludes that the R-22 system is the more cost effective option. Of course, for a specific application, a detailed economic investigation should be undertaken before the selection of any one system approach.

DESCRIPTION OF THE NCSU HEAT PUMP SYSTEM

A layout of the heat pump system as it was first constructed is shown in Figure 7. Primary considerations in the selection of the equipment shown were cost, availability, portability, and compatibility with the wind machine output characteristics. Portability was necessary due to the requirement of transporting the system to New Mexico for participation in the SCORE competition. The last factor stated above stemmed from the need for the overall system to be energy self-sufficient. This resulted in the requirement that the compressor be externally driven, the output of the wind machine being compressed air used to drive an air motor.

During initial testing of the system, the wind machine was replaced by a .746 kW electric motor to drive the compressor. Further modifications to the original system and a brief description of each follows:

(1) The thermostatic expansion valve malfunctioned and was replaced by a manual throttling valve to afford greater control of refrigerant flow to the evaporator.

(2) An oil separator was added to the compressor discharge line to aid in oil return to the compressor.

(3) The condenser was removed from the water storage tank and replaced by an external shell and tube condenser, the water being pumped in a closed loop between the storage tank and the condenser.

(4) The liquid receiver was removed since excess refrigerant could be stored in the condenser.

(5) A suction line accumulator was installed to protect the compressor from possible liquid carryover from the evaporator during quickly fluctuating loads.

(6) The motor power was changed to 1.12 kW and compressor speed was increased in an effort to boost system capacity. Eventually, the belt driven automobile compressor-electric motor combination was replaced by a 3.73 kW hermetically sealed heat pump compressor.

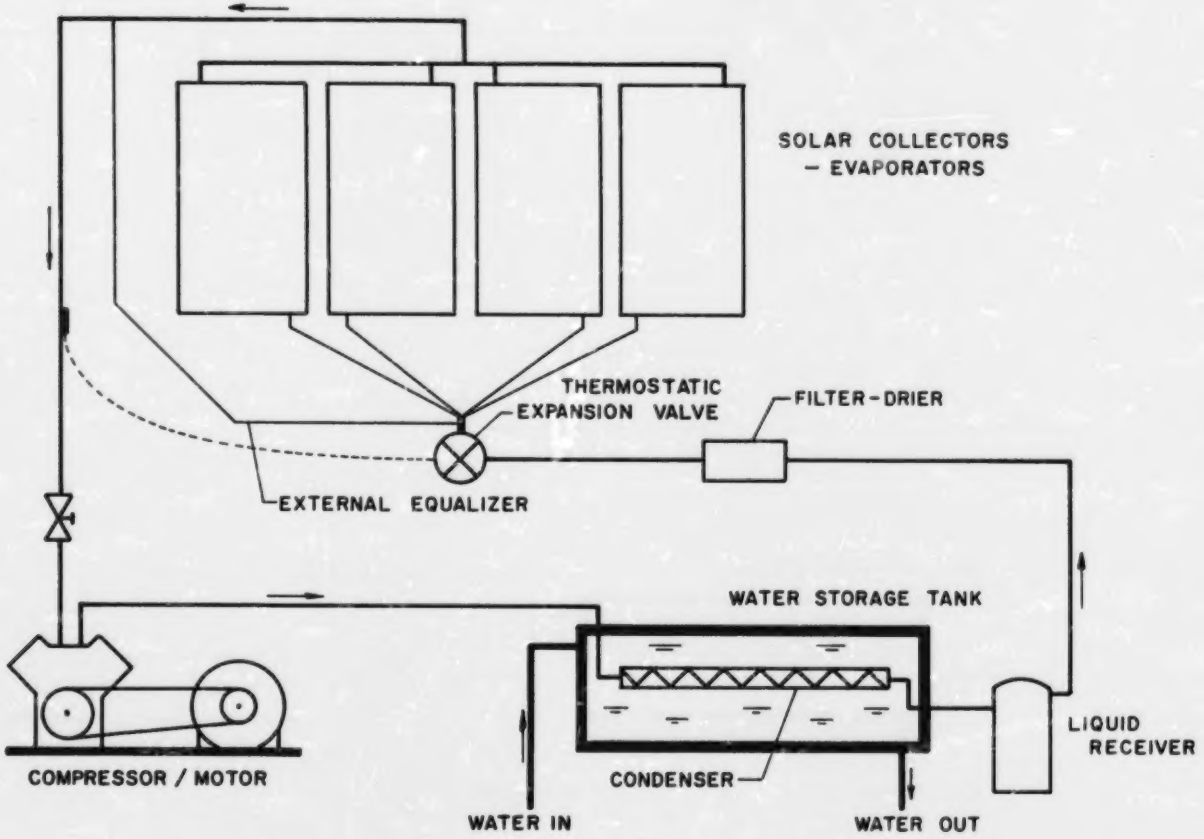


FIGURE 7. Initial System Layout.

(7) A larger shell and tube condenser was installed in the system in an effort to reduce condensing temperatures.

A schematic of the final system layout is shown in Figure 8. More detailed descriptions of some of the principal system components are given below.

1. Solar Collector-Evaporator

The solar collector-evaporator system consists of four units, each of about 2.044 square meters of a combined total of 8.176 square meters collector area. Construction is of 5.08 cm x 10.16 cm wood framing. Polyurethane foam insulation was sprayed behind the absorber plate to a nominal thickness of about 5.5 cm. The absorber plate is an aluminum Roll-Bond panel manufactured by Olin Corporation (Model RB-7727). Details of the flow distribution and the dimensions of the flow passages are shown in Figure 9. The plate is covered with an experimental "paintable" selective coating consisting of PbS precipitate mixed with an activated silicone resin (Dow Corning paint E1-9752, Resin 825, 55 wt% PbS). Glazing for the collectors is 4 mil "Tedlar" PVF film (DuPont 400BG20TR). The film transmits about 92-94% of the total incident solar energy, with main losses caused by surface reflection. It is slightly superior to glass in short wave transmittance, but is not as nearly opaque to the long wave radiation emitted by typical flat plate absorber panels. Selection was primarily based on considerations of weight and damage resistance.

2. Compressor

The 3.73 kW compressor, selected on a basis of cost and availability was necessary to provide additional system capacity. It is a standard

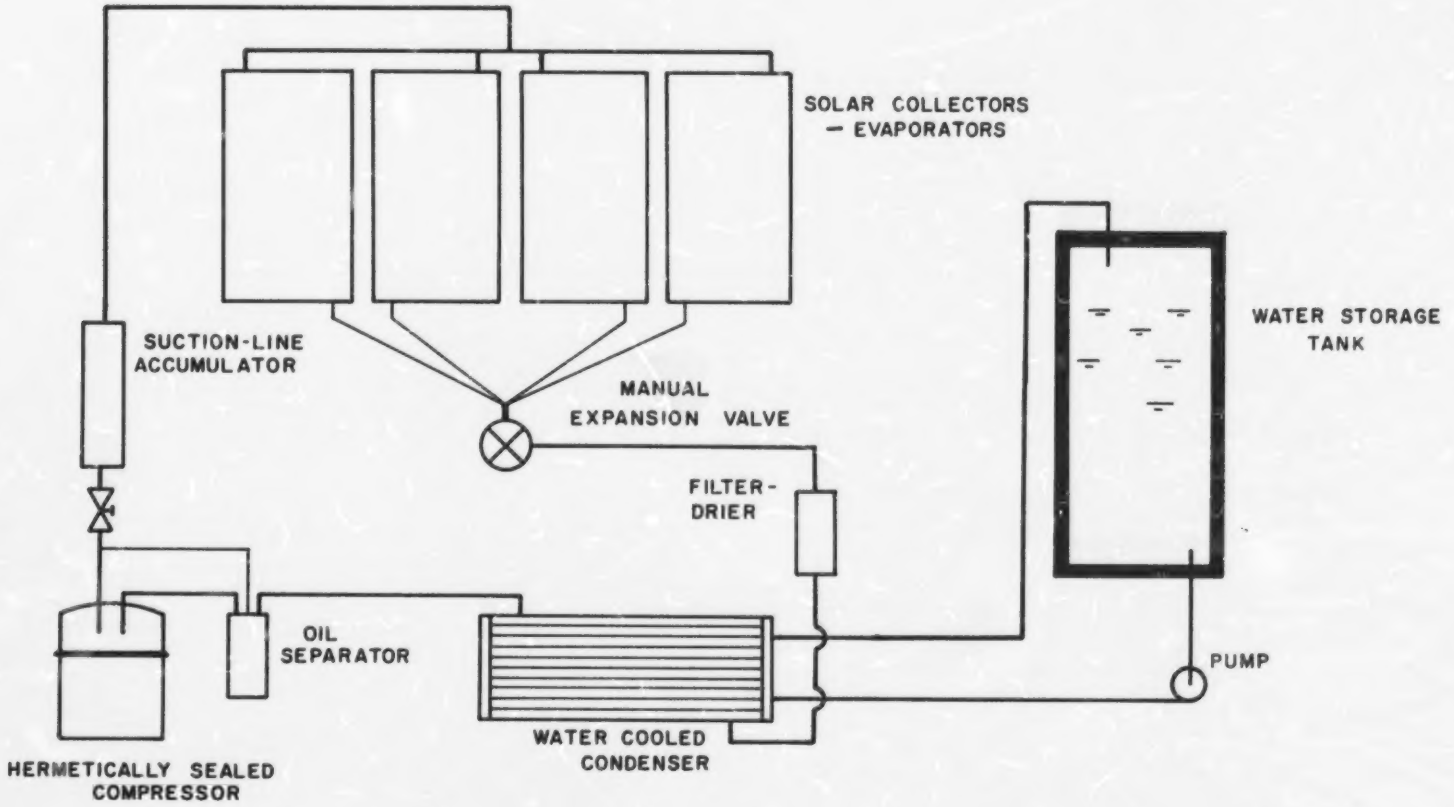


FIGURE 8. Final System Layout.

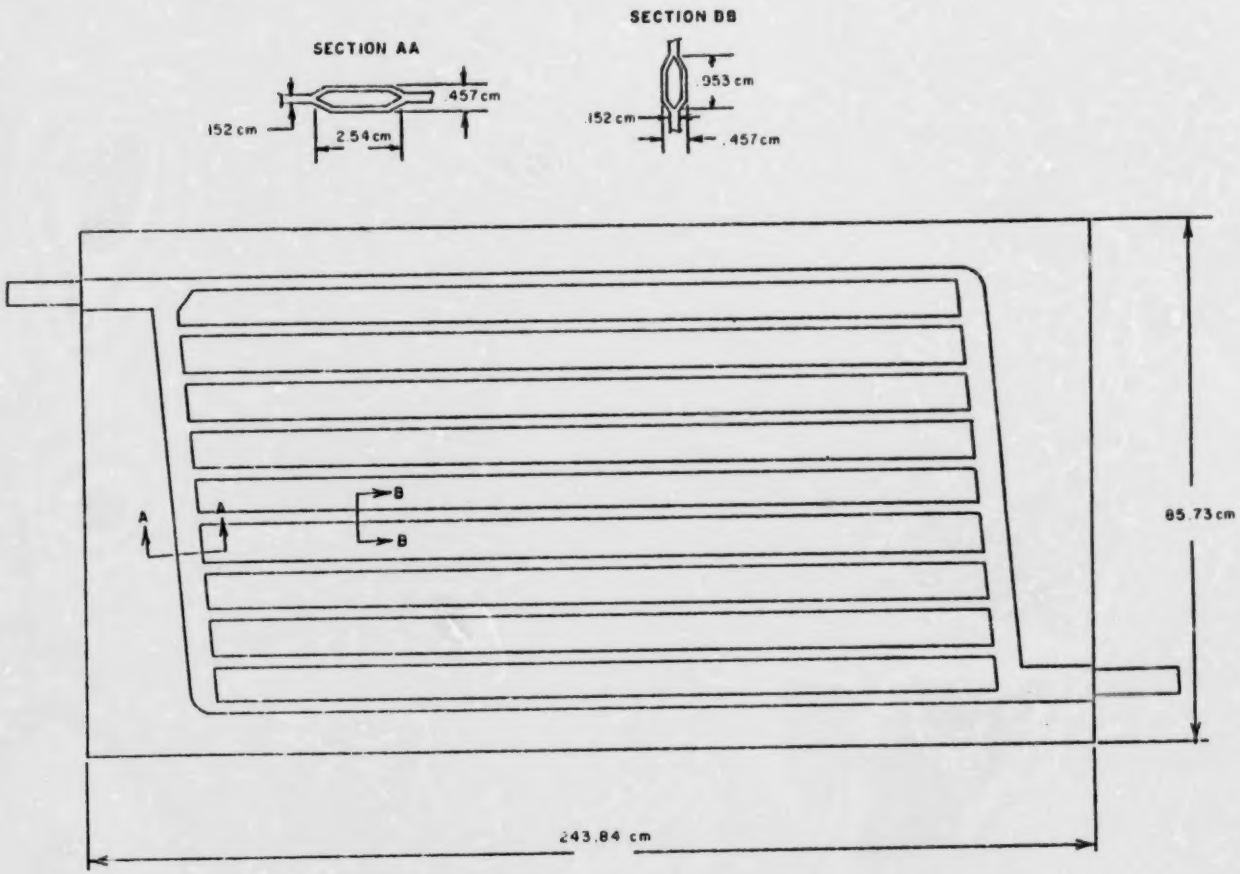


FIGURE 9. Collector Plate Detail.

heat pump compressor manufactured by Copeland (Model YRE4-0500-PFB). The unit is hermetically sealed and designed for R-22 refrigerant. Compressor displacement is 24.3 m^3/hr. Cooling is provided by suction gas entering the compressor.

3. Refrigerant

The refrigerant selection was based on the following general considerations: (1) compatibility with system components at operating conditions; (2) cost and availability; (3) compatibility with common lubricants; and (4) safety. Due to structural limitations the maximum allowable working pressure in the evaporator is 5.52 bars gage. Thus, the refrigerant pressure must not exceed this limit at the expected operating temperatures. This factor immediately eliminates the use of the more common refrigerants R-12 and R-22. However, since most readily available refrigeration components are designed for these refrigerants, the refrigerant chosen must possess characteristics as similar to R-12 and R-22 as possible.

The refrigerant chosen to satisfy these requirements is R-114 ($CCIF_2$ - $CCIF_2$). In addition to meeting the pressure conditions stated above, its physical properties (excluding density and boiling point) very nearly parallel those of R-12 and R-22, thereby making the use of components designed for these refrigerants possible if suitable sizing requirements are fulfilled.

PERFORMANCE TESTING PROCEDURE

The overall objective of the performance test was to obtain sufficient data to determine complete energy balance of the system operating under various conditions. The evaluation of system performance was made following accepted refrigeration practice and solar energy standardization procedures. Accordingly, the results are presented in terms of the heat pump coefficient of performance (COP) and the solar energy extraction efficiency as they vary under different combinations of solar intensity, ambient temperatures, and condensing temperatures. The system layout during the performance testing was essentially as shown in Figure 8 with the exception of a heat sink provided by a heat exchanger mounted in a forced air duct. The coil was located between the circulating pump and the storage tank, piping being provided by rubber hoses. The fan in the duct system could be operated intermittently as the need for a heat sink was indicated.

In order to generate the desired outputs stated above, the relevant parameters measured and/or recorded are as follow:

(1) ambient temperature, (2) general atmospheric conditions, (3) incident solar flux (beam plus diffuse) normal to the collector surface, (4) electrical energy required to drive the compressor, (5) cooling water flow rate, (6) cooling water temperatures at condenser inlet and outlet, (7) high side (condenser) pressure, (8) low side (evaporator) pressure, (9) suction temperature at the compressor inlet, (10) discharge gas temperature at the compressor outlet, (11) discharge gas temperature at the condenser inlet, (12) liquid temperature at the condenser outlet, (13) liquid temperature at the expansion valve inlet, (14) liquid-vapor temperature at the evaporator inlet, (15) suction gas temperature at the evaporator outlet, (16) temperatures of the lower, middle, and upper portions of the absorber panel, and (17) refrigerant flow rate.

Power required to drive the circulating pump and heat sink fan was not measured since only the performance of the heat pump itself is under consideration. A brief description of the instrumentation employed for data collection is given below.

1. Solar insolation: The solar radiation (beam plus diffuse) was measured with an Eppley Black and White pyranometer model 8-48. Readout instrumentation was provided by a Leeds and Northrup strip chart, millivolt potentiometric recorder model 69800. Standardization of the pyranometer was furnished by Eppley.

2. Temperature: Temperatures throughout the system were measured by iron-constantan thermocouples suitably attached at the desired locations. Readout was provided by a Leeds and Northrup Speedomax G strip chart, multipoint potentiometric recorder model 60362. Calibration was established through the use of ice bath and boiling water constant temperature sources.

3. Flow rates: Cooling water flow rate was measured by a suitably calibrated rotometer. Refrigerant flow rate was measured by a Fisher and Porter variable area flowmeter model M1-1186/1 designed for R-12 at a specific gravity of 1.29. Calibration of the instrument for R-114 was not suitably established. Consequently, determination of the refrigerant flow rate was made through a heat balance on the condenser.

4. Pressure: The system pressures were measured with Bourdon-tube gauges fixed in the piping system and by a standard refrigeration test manifold attached at the compressor inlet and discharge lines.

5. Electrical energy: Electrical energy required to drive the compressor was measured by a standard voltmeter, ammeter, and wattmeter.

Data were generally recorded in one-half hour increments, except as noted in the remarks above. While the bulk of data were taken on sunny days, testing was also performed during unfavorable conditions. This was done to establish lower operating limits and to obtain time lag characteristics of the system when operating under rapidly fluctuating loads.

RESULTS OF HEATING MODE PERFORMANCE TESTING

Data taken during the heating mode performance testing is given in Appendix 1. The data included are limited to those taken while the system was operating in a relatively steady state manner. System operating characteristics for selected days are shown graphically in Figures 10 and 19. Sky conditions were clear except for October 30th when the sky was completely overcast. Note the scale displacement in Figure 16.

The relatively high condensing temperatures for the July 16th and August 12th test dates were due to the inadequate surface area of the smaller condenser. The remaining data shown were taken after the installation of the larger condenser. However, it was found that the heat sink arrangement did not function as hoped due to excessive pressure drop in the heat exchanger. Therefore, the cooling water was circulated in a closed loop from the storage tank to the condenser. Thus, as energy was added to storage a rise in condensing temperatures resulted and system operation eventually had to be discontinued when the condensing temperature reached about 80^{o}C.

The solar collector efficiency presented in this report is defined as the ratio of useful heat gain from collector to the incident solar flux normal to the collector plate. As can be seen from the graphs, the efficiency remained very high for all four days. In fact, efficiencies of greater than 100% occurred if evaporating temperatures dropped below the ambient temperature as on October 30th. This would be expected as the collector would also be absorbing heat from the surrounding air.

While the typical coefficient of performance (COP_h) value of about 3 is not unacceptable, it is below expectations. It is felt that this

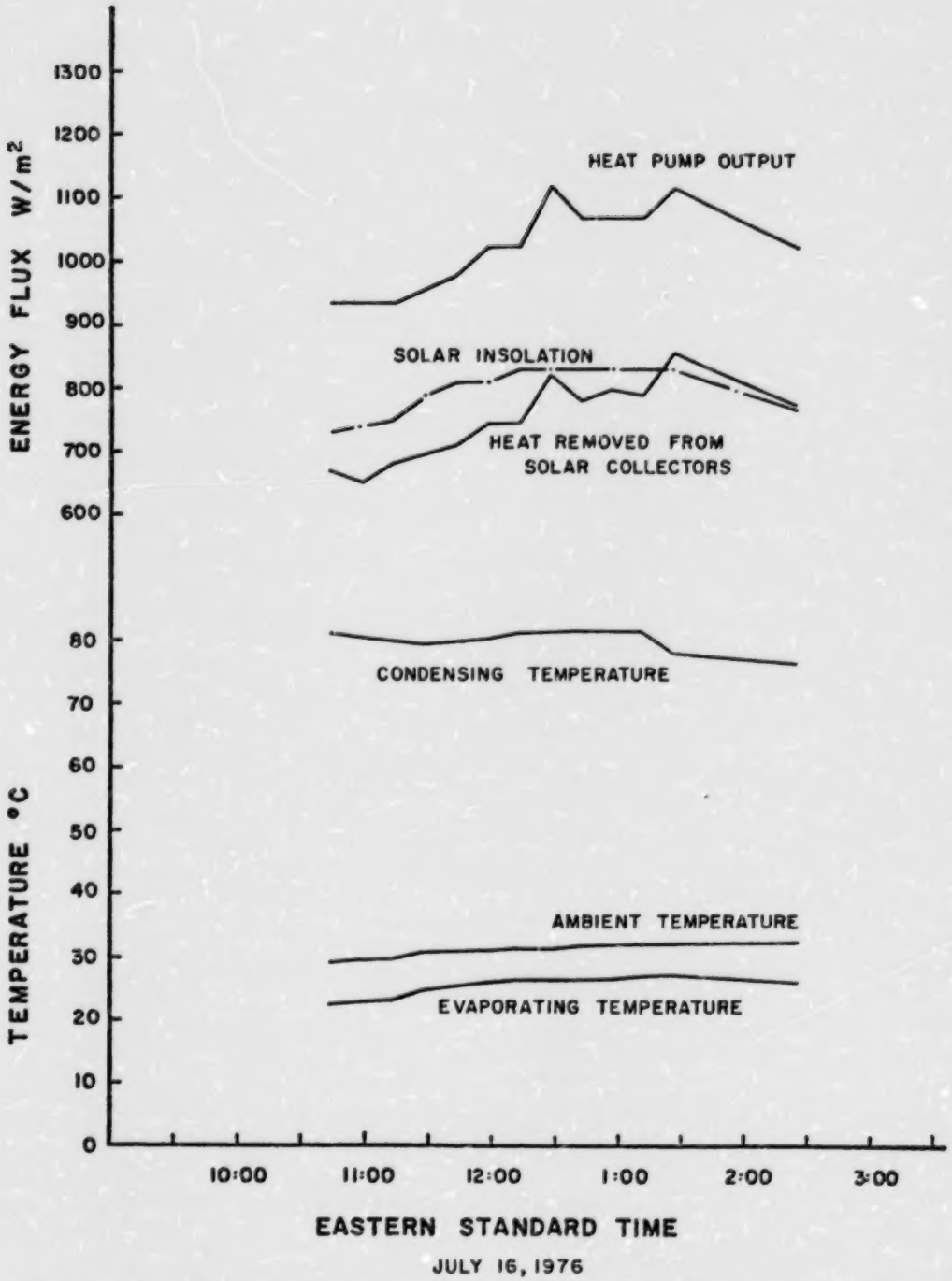


FIGURE 10. Heating Mode Performance Test Results.

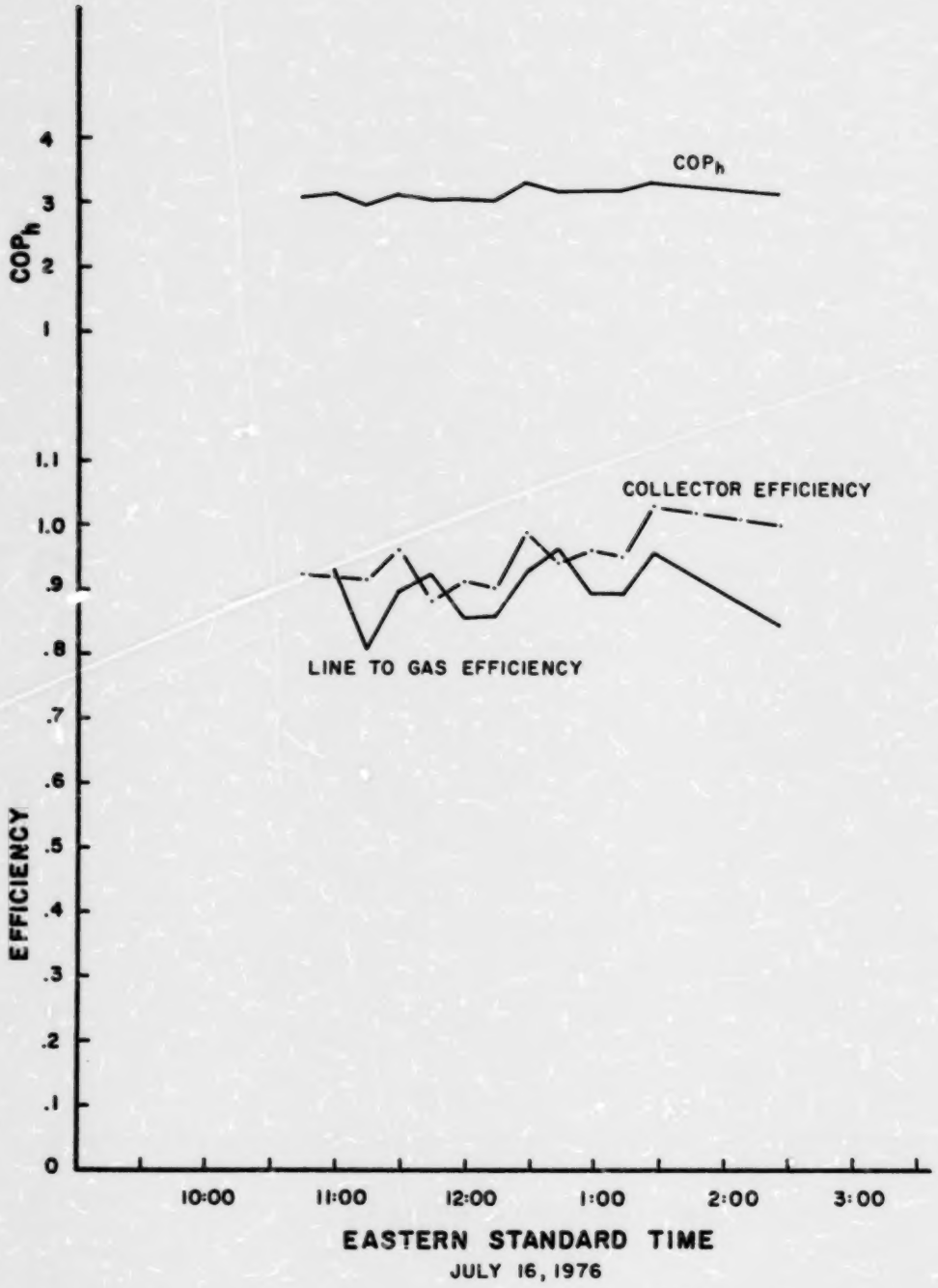


FIGURE 11. Heating Mode Performance Test Results.

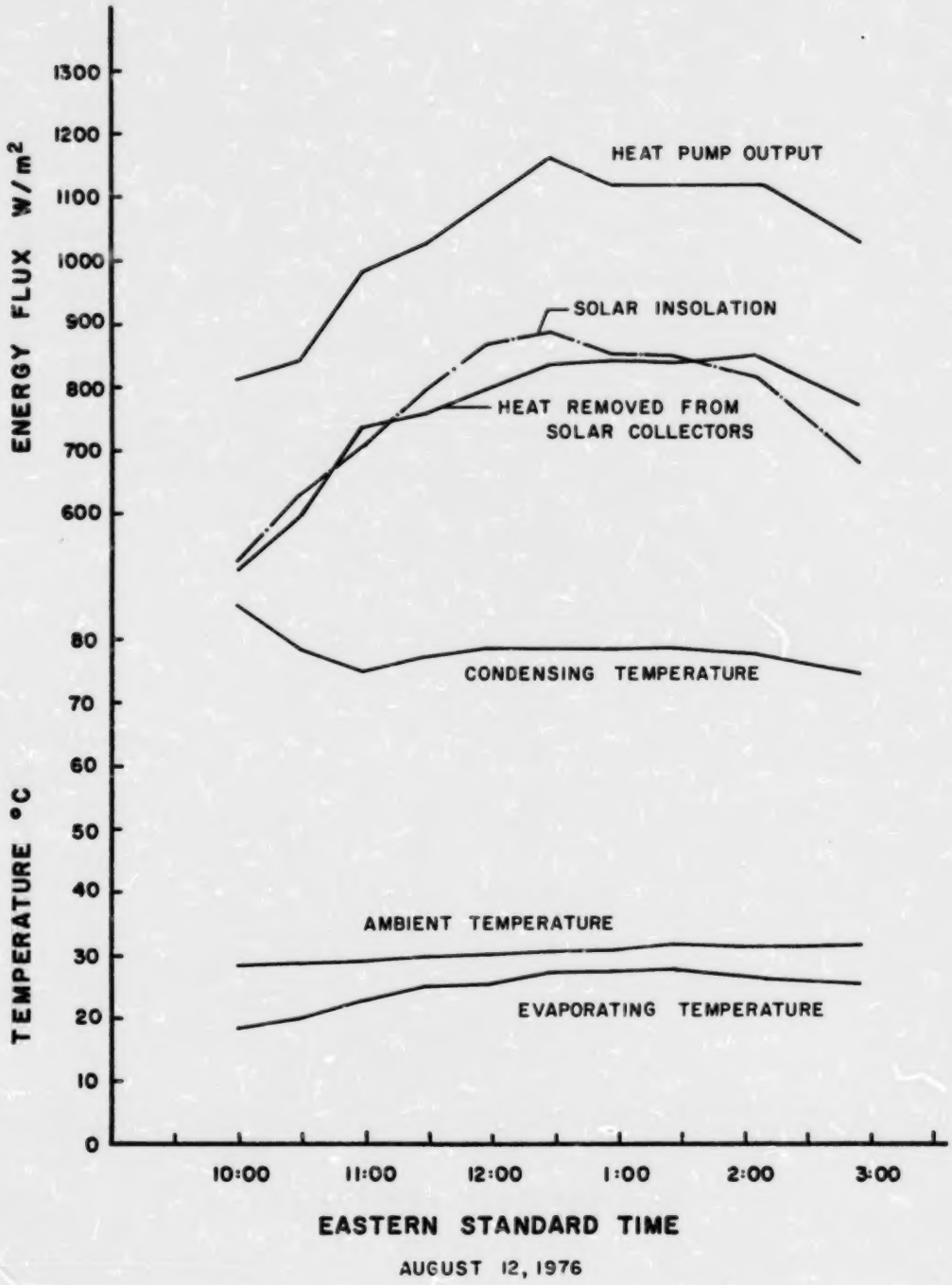


FIGURE 12. Heating Mode Performance Test Results.

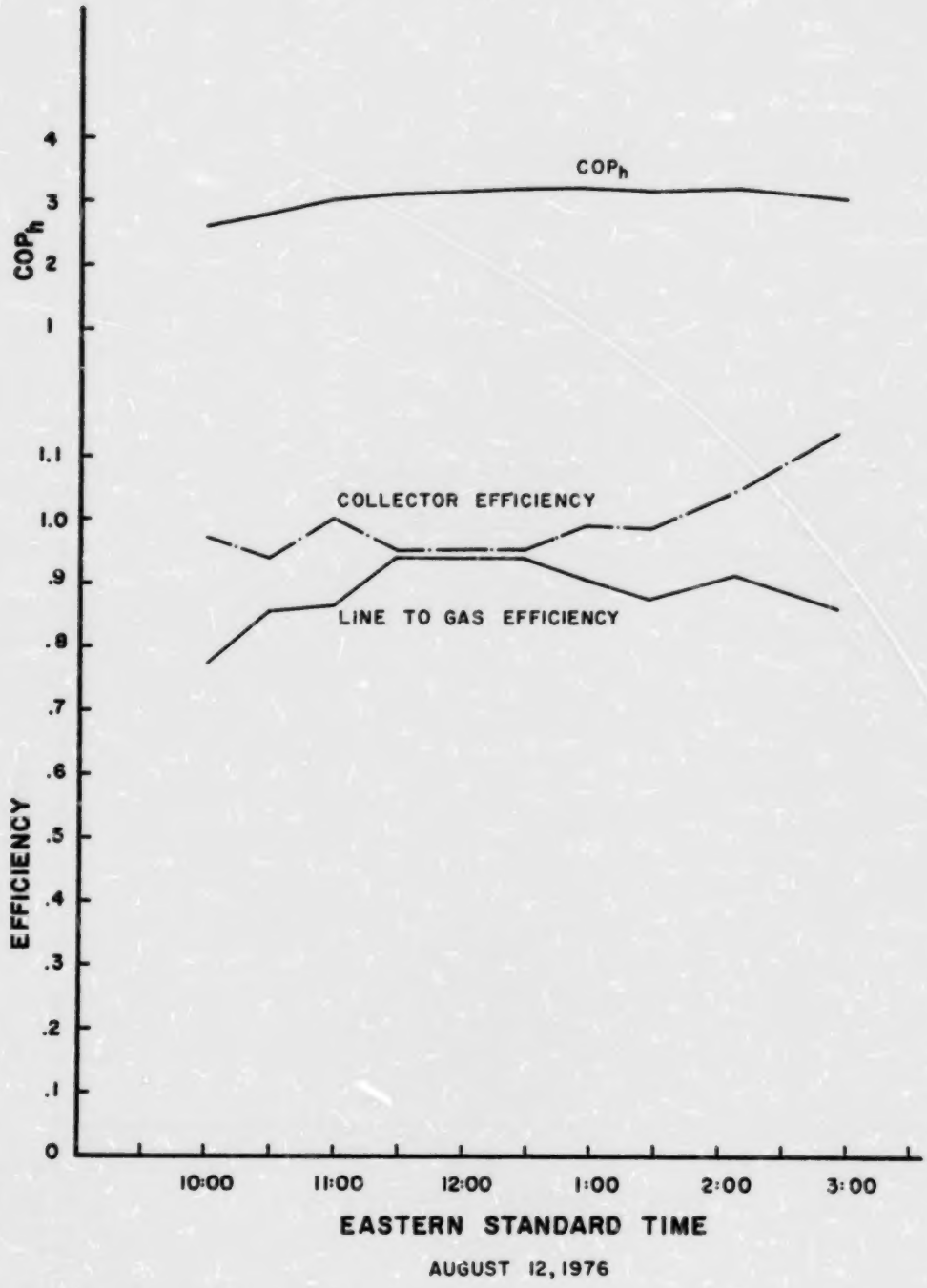


FIGURE 13. Heating Mode Performance Test Results.

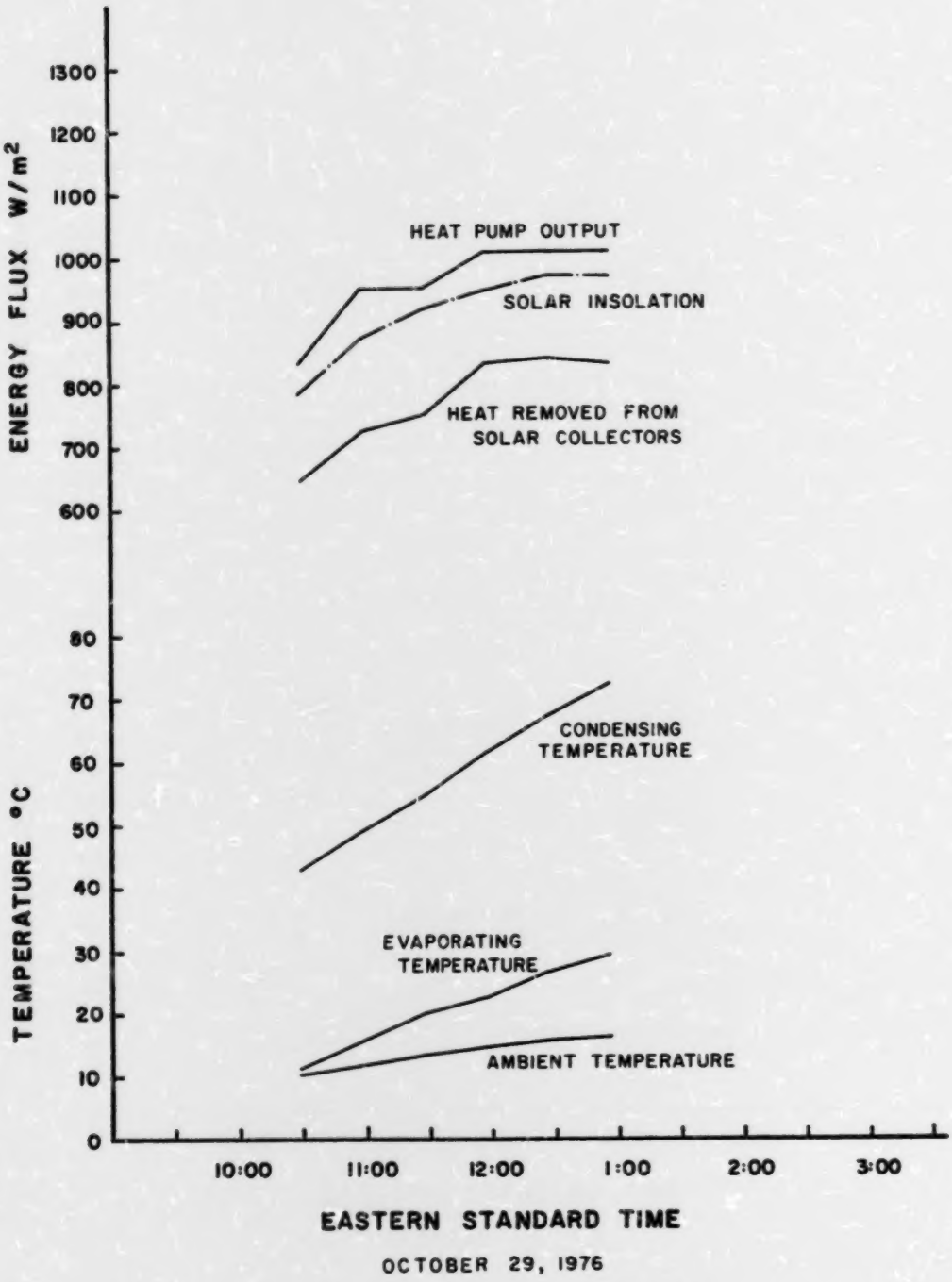


FIGURE 14. Heating Mode Performance Test Results.

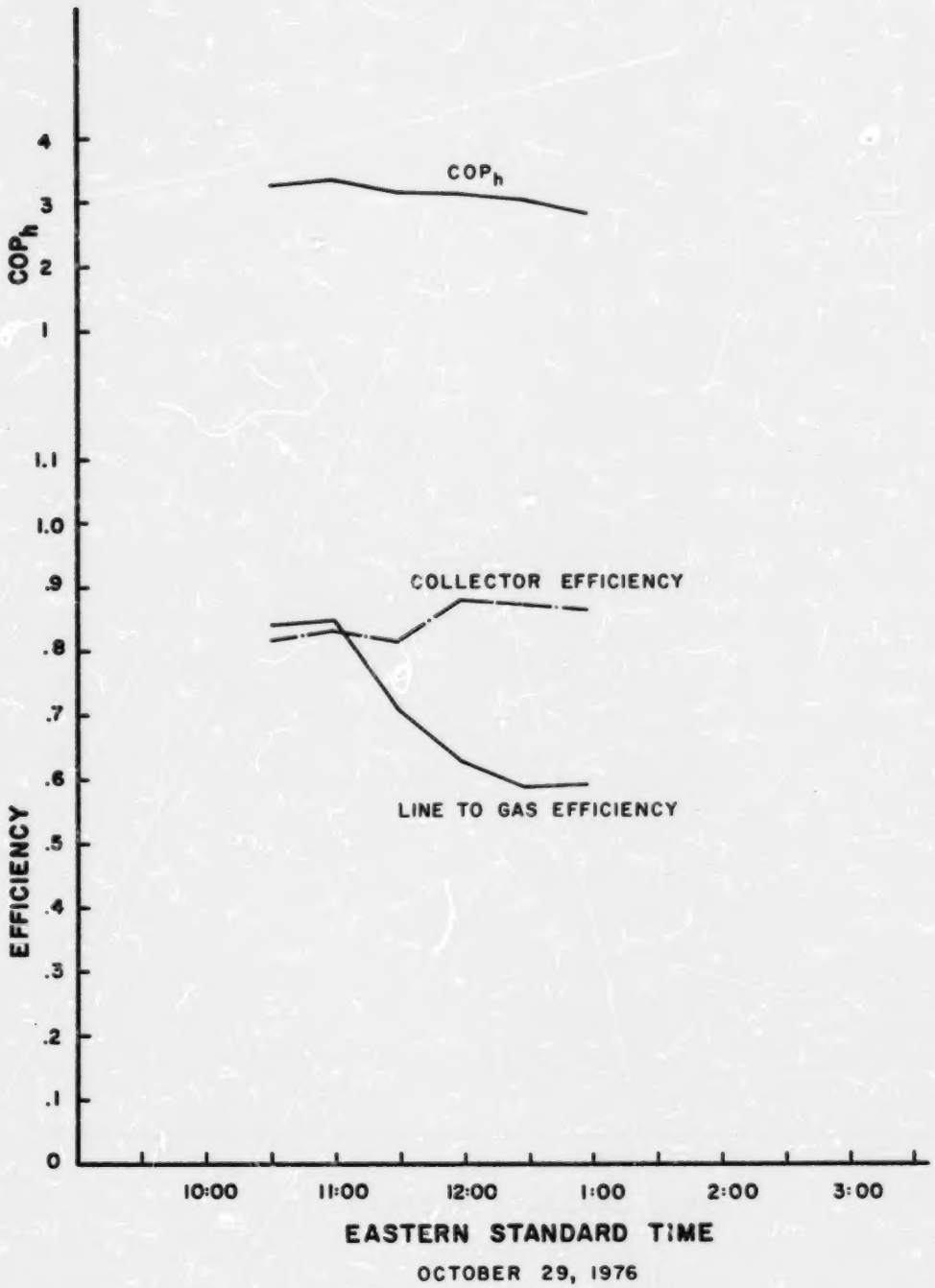


FIGURE 15. Heating Mode Performance Test Results.

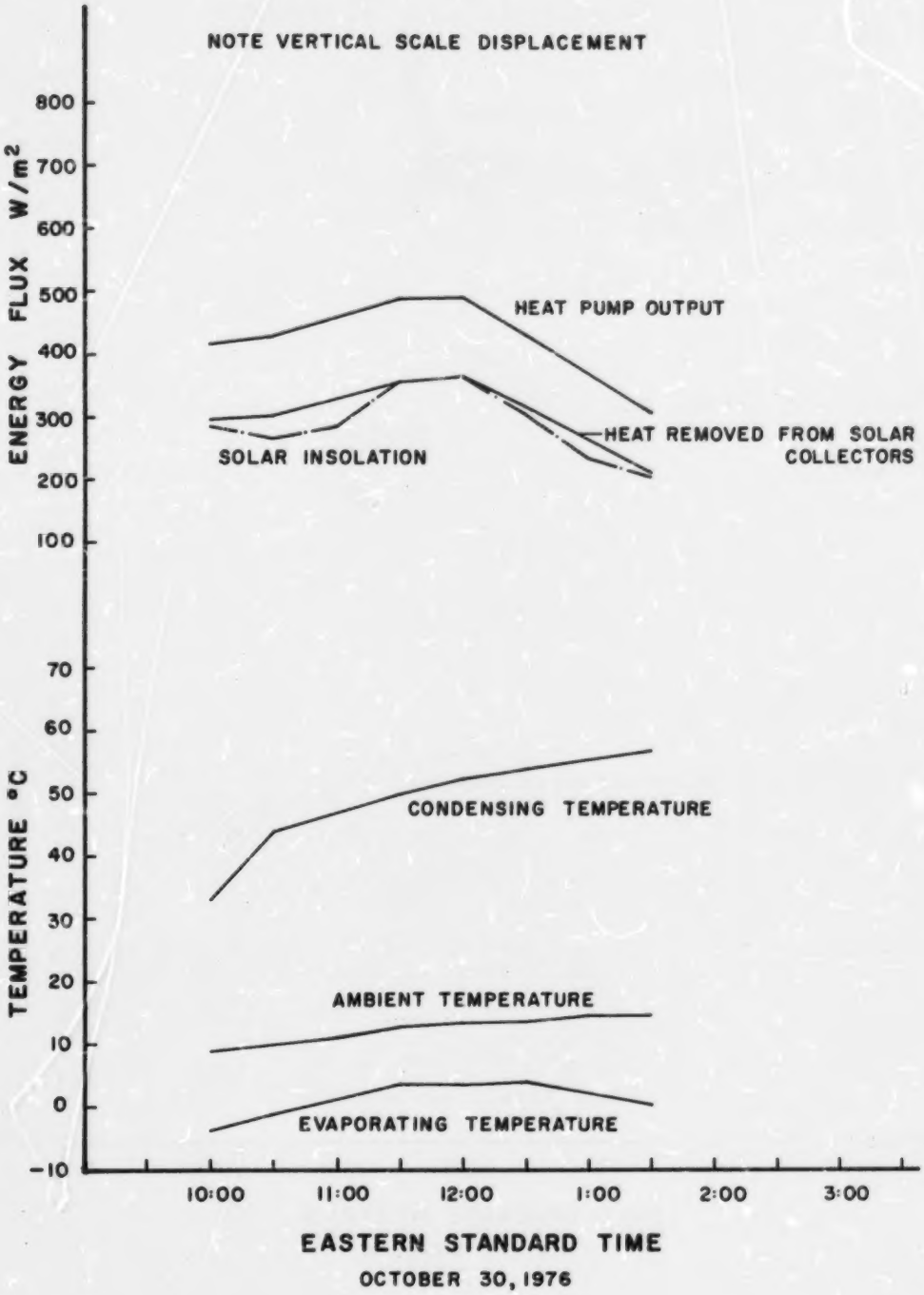


FIGURE 16. Heating Mode Performance Test Results.

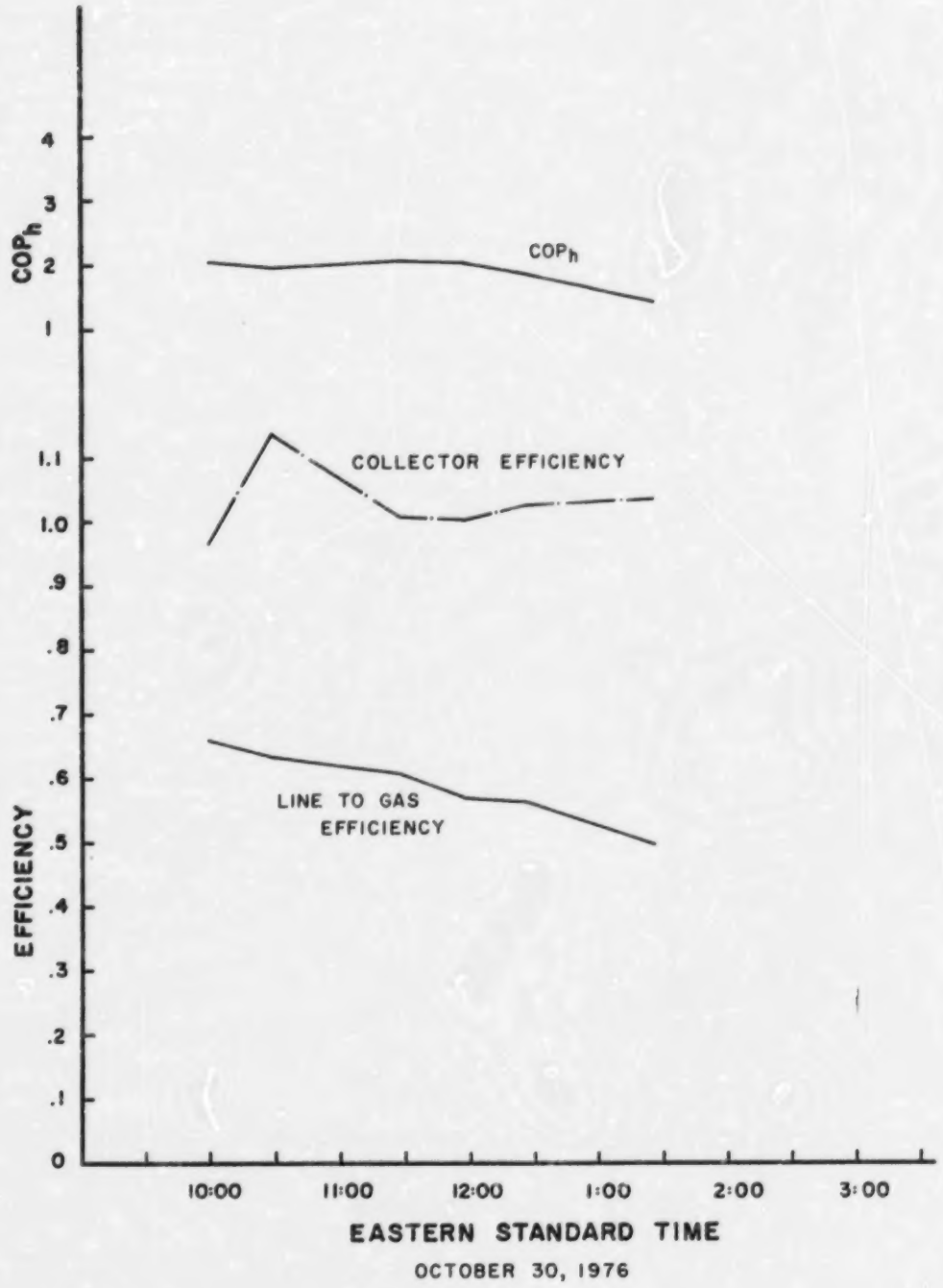


FIGURE 17. Heating Mode Performance Test Results.

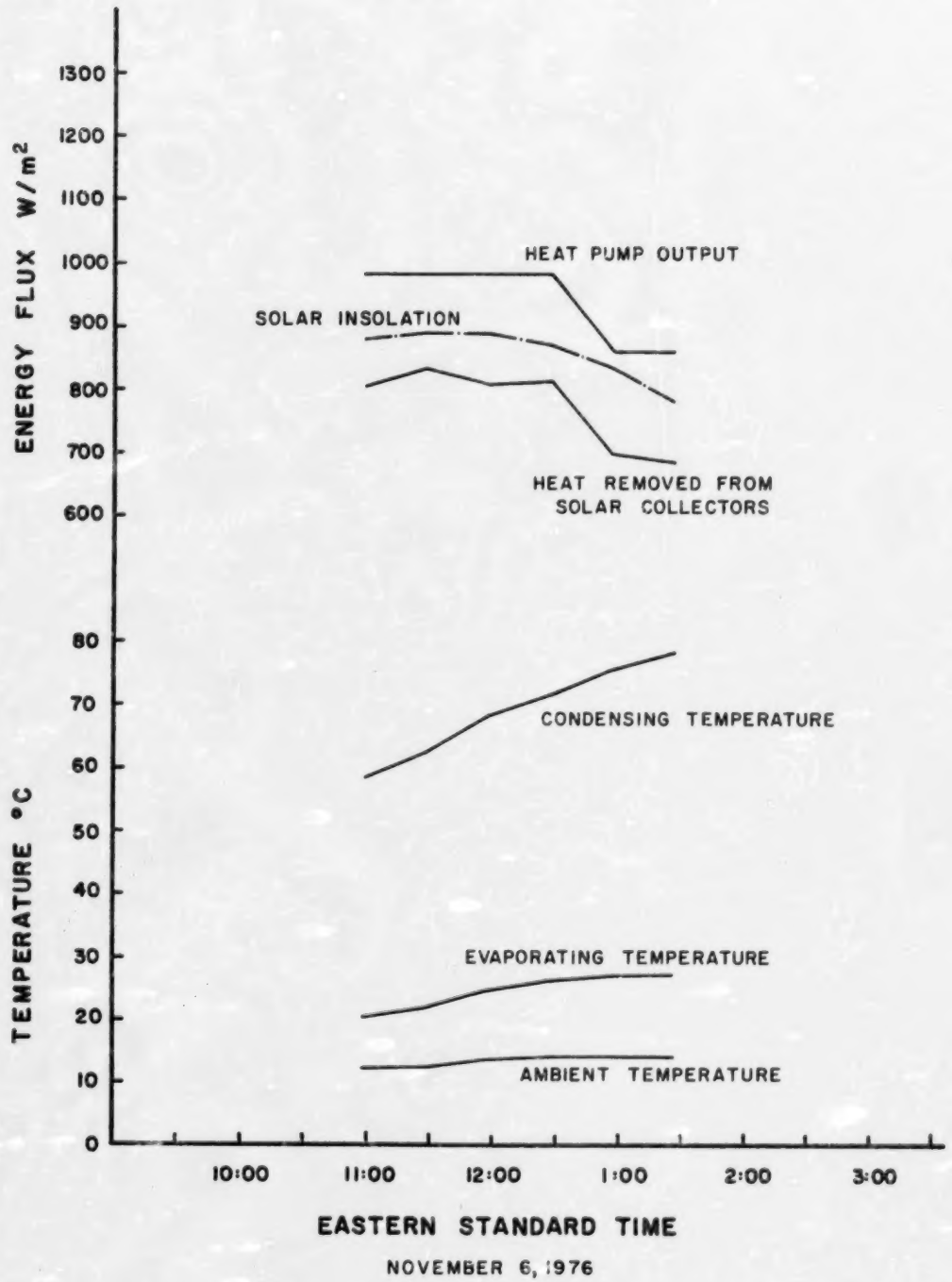


FIGURE 18. Heating Mode Performance Test Results.

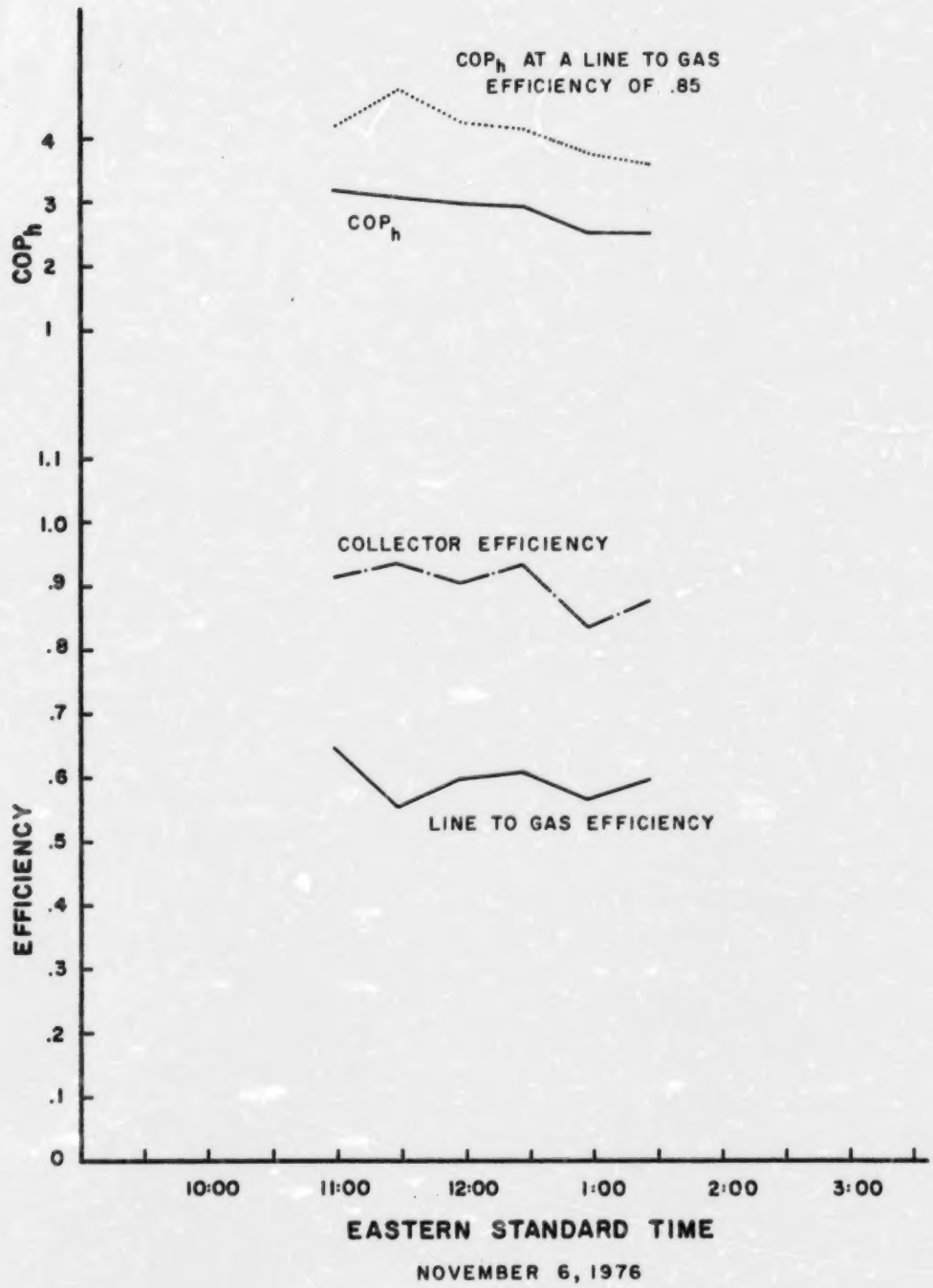


FIGURE 19. Heating Mode Performance Test Results.

is a result of a poor refrigerant - compressor - motor match, leaky valves and piston rings in the compressor, and inadequate capacity control.

As discussed in previous sections, the solar collector plate required the selection of a relatively low pressure refrigerant. But most readily available refrigeration equipment is generally designed for R-12, R-22, or R-502. An externally driven compressor was found to be beyond budget restrictions and therefore, a R-22, hermetically sealed compressor was selected on the basis of the displacement requirements of R-114 under expected maximum load conditions. Since R-22 is a more dense refrigerant the motor power capability of the sealed compressor unit was oversized, resulting in somewhat poorer performance. This effect is shown more clearly in Table I which tabulates theoretical operating characteristics for one ton of heat removed from the evaporator. Note that for standard air conditioning temperatures of 27.2°C - 54.4°C a five ton, R-22 machine would displace about 17.24 m^3/hr and require 4.18 kW. This displacement would correspond to that required for R-114 at temperatures of 25°C and 60°C at a load of a little over two tons. However, the power necessary for the R-114 machine at this operating condition would be only .925 kW or less than one-fourth the power available. Thus, the motor would operate at much below its rated capacity.

It is felt that the COP_h was also adversely affected when the system operated at reduced loads. Since the only capacity control on the system was the manual expansion valve, no means of altering compressor displacement was available. This resulted in lower evaporating pressures and a subsequent increase in power requirements and drop in the coefficient of performance.

TABLE I. Comparison of Operating Characteristics for Refrigerants R-22 and R-114.

Evaporating Temperature °C	-15		4.4		27.2		25	
Condensing Temperature °C	30		40.5		54.4		60	
REFRIGERANT	R 22	R 114	R 22	R 114	R 22	R 114	R 22	R 114
Refrigerating Effect $\frac{kJ}{kg}$	162.8	101.6	156.5	101.1	138.3	96.7	135.7	99.9
Theoretical Liquid Circulated $\frac{kg}{ton\ min}$	1.30	2.20	1.35	2.10	1.52	2.18	1.55	2.11
Theoretical Compressor Displacement $\frac{m^3}{ton\ hr}$	6.03	32.77	3.31	15.56	3.45	14.71	2.11	8.31
Theoretical Compressor Power Requirement $\frac{kW}{ton}$	.768	.858	.553	.538	.835	.680	.524	.446

Leaky valves and piston rings in the compressor are felt to be responsible for the rather dramatic decrease in line-to-gas efficiency of the compressor unit. Line-to-gas efficiency is taken to be the ratio of the enthalpy change of the refrigerant across the compressor to the electrical work input. Efficiencies of about 85% to 90% were typical when the unit was relatively new while values dropped to about 60% by the end of the performance testing. Excessive discharge temperatures and inadequate oil return are the most likely causes for the leakage. If a line to gas efficiency of 85% had been maintained during the entire testing period, the COP_h would have been considerably improved. For example, on November 11th the COP_h would increase to the dotted lines shown in Figure 19. The mean value of about 4.0 is considered well within the range attainable with more refined system design. Recommendations with regard to insuring adequate oil return and decreasing discharge temperatures are given later in this report.

As was mentioned earlier, some data were taken while operating the system under rapidly fluctuating loads. This was done primarily to establish time lag characteristics for use in the selection of refrigerant flow and system capacity controls. Figure 20 shows changes of upper panel and compressor inlet temperatures for a period of highly variable solar intensity. The damping effect is due to the suction line accumulator which functions as a thermal capacitance while fulfilling its primary role of protecting the compressor from liquid carryover. A discussion of recommended refrigerant flow controls is given in a later section.

It should be noted here that while care was taken to insure as accurate measurements as possible, budget and time restraints required some compromises. Instrument selection was generally restricted to what

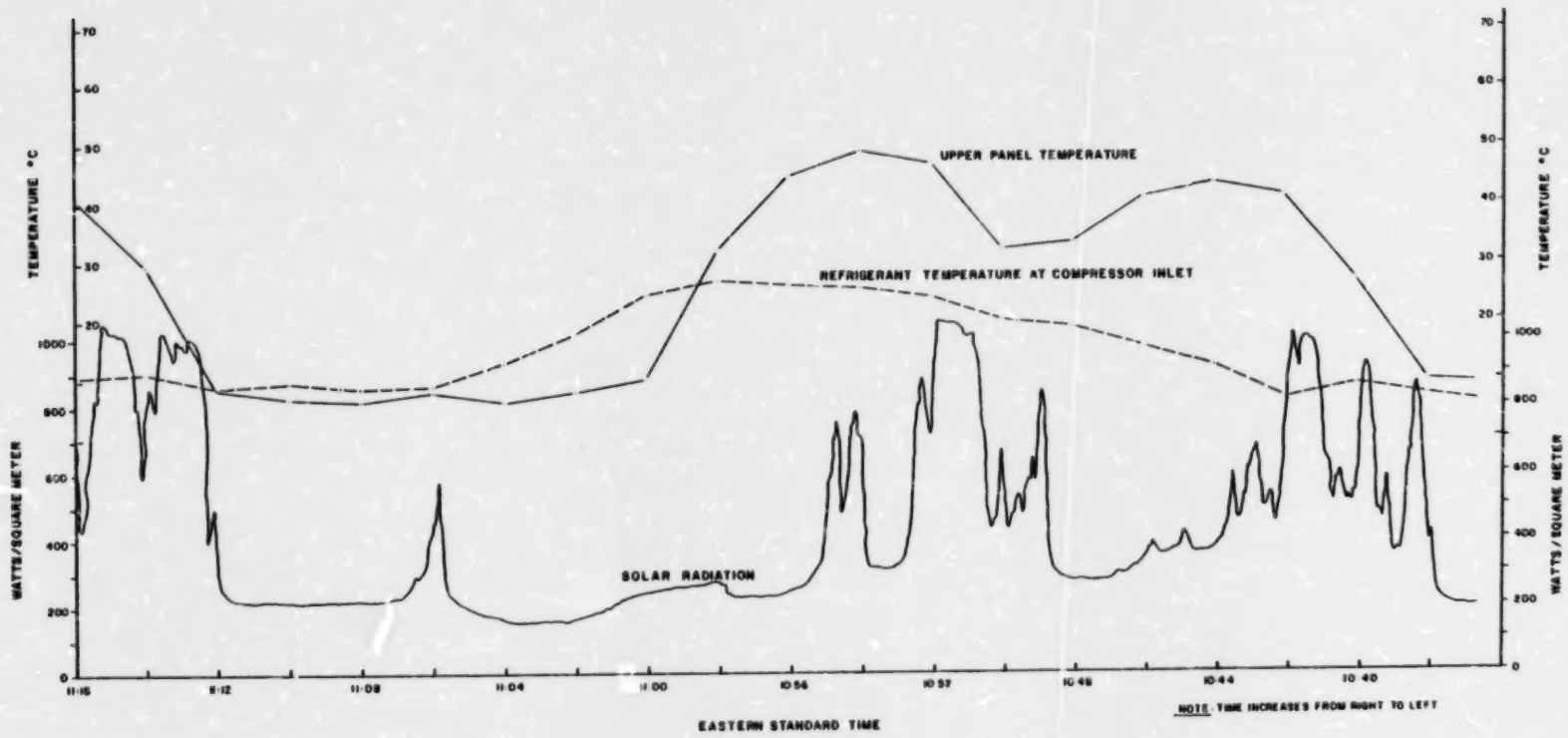


FIGURE 20a. System Response to Quickly Fluctuating Solar Intensity.

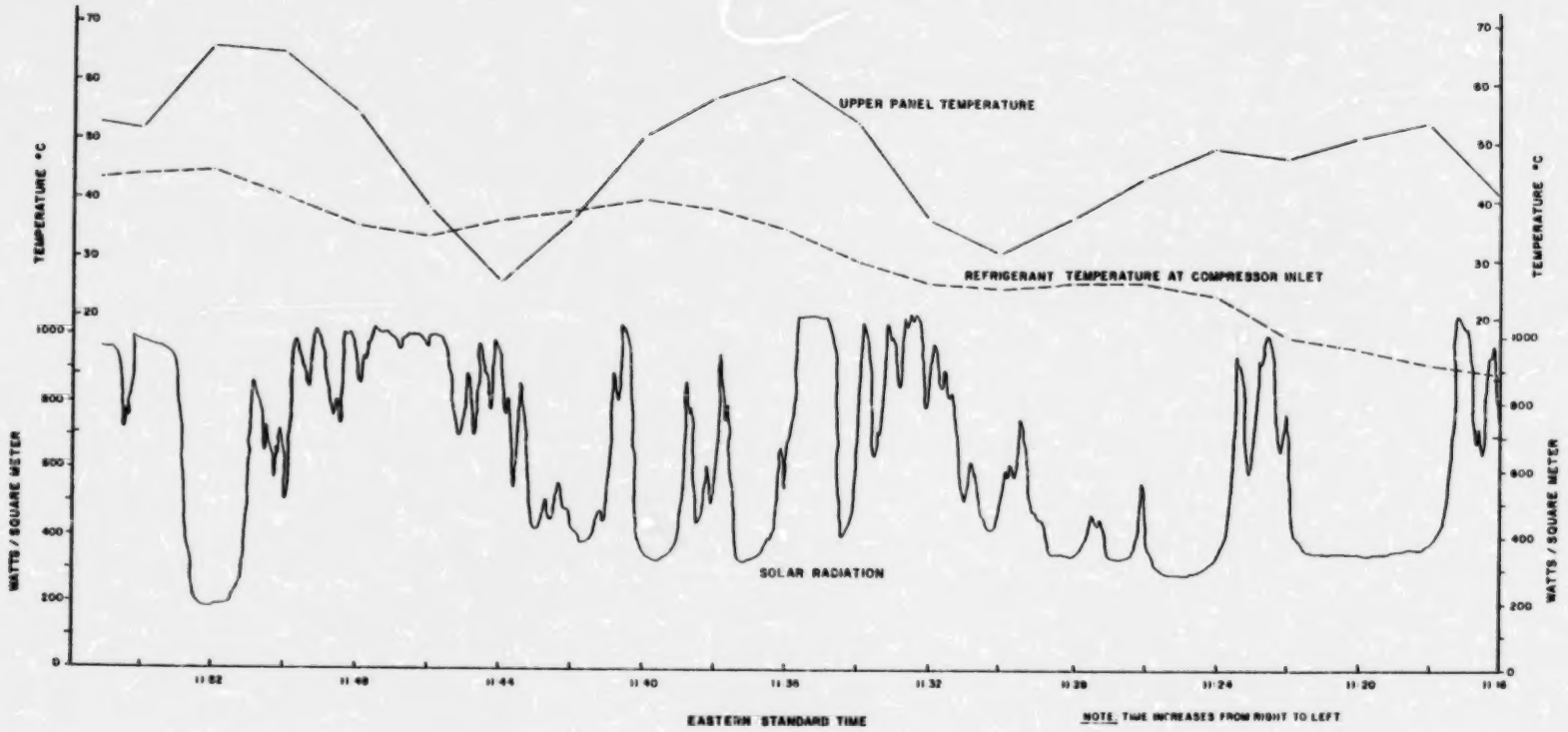


FIGURE 20b. System Response to Quickly Fluctuating Solar Intensity.

was on hand in the department, the only notable exception being the pyranometer. Therefore, due to instrumentation inaccuracy, the data presented should be regarded as qualitative. It is felt, however, that the soundness of the basic principal of the solar assisted heat pump concept is clearly demonstrated. That is, an improvement of both solar collection efficiency and heat pump coefficient of performance can be realized through the use of such a system.

DESIGN CONSIDERATIONS AND RECOMMENDATIONS

Heating cycle operation of the heat pump system presented several problems that perhaps are not normally encountered in more conventional refrigeration systems. The most outstanding of these was compressor overheating, which eventually led to valve failure in the belt driven automobile compressor initially used in the system. This problem, while aggravated by the fact that heating mode operation was being performed during the summer months, was primarily a result of inadequate compressor capacity and poor lubrication. The insufficient capacity stemmed from an underestimation of the effectiveness of the solar panels at the design stage. The predictable outcome was substantial superheating of the suction gas, resulting in excessive discharge temperatures.

The high compressor temperatures also contributed to lubrication problems as more oil was carried out by the refrigerant than would normally be expected. Had the oil been circulated throughout the system and returned to the compressor, little damage would have occurred except reduced effectiveness of condenser and evaporator surfaces. However, while the system was pumped down for expansion valve replacement, it was found that a considerable amount of oil had become trapped in the lower portion of the solar panel evaporator. Ideally, the evaporator tube sizing should be small enough to insure adequate refrigerant velocity to carry the oil to the suction line and the compressor. Practically, however, this condition is not always obtainable due to the large vertical rise of the collectors and the relatively large range of flow rates required to meet fluctuating load demands. Extended operation of the system therefore resulted in inadequate compressor lubrication, which further

added to the overheating problem and the eventual valve failure.

In an effort to overcome these difficulties, a larger motor was obtained to drive the compressor and an oil separator was installed at the compressor discharge. Additionally, compressor speed was increased and a higher viscosity oil was secured. Results of these modifications were generally favorable, although system capacity was still not adequate, probably attributable to leaky compressor valves and piston rings. The oil separator functioned fairly well, but refrigerant condensed in it during the off cycle. This required manually throttling the oil return line to prevent liquid flood-back to the compressor during start-up. Further complications arose over long term operation since the separator was not 100% efficient and oil eventually became trapped in the evaporator once again.

As more funds became available, a 3.73 kW hermetically sealed R-22 compressor was obtained and installed in the system. The oil separator was placed on the discharge line as before, the oil return line being joined to the suction line just upstream of the compressor. This change resulted in more than adequate capacity and the system performed reasonably well, although not up to expectations (as discussed in another section of this report). However, toward the end of the testing period the compressor had become considerably more noisy and somewhat less efficient, again, probably due to loss of oil and/or valve and piston ring leakage.

Several possible solutions exist with regard to the oil return problem. During the off cycle, the oil and liquid mixture in the evaporator may be drained to the accumulator. Upon start-up, the oil and liquid refrigerant mixture would then be metered to the suction line. This

alternative has the disadvantage of requiring additional piping and controls and a significantly larger accumulator. Vapor leaving the evaporator would also have to be superheated to a greater extent as it would give up heat to the liquid stored in the accumulator.

Another and perhaps more promising solution to the oil return problem would be to operate the system with a flooded evaporator. This alternative would materially shorten the length of travel of the vapor borne oil in the evaporator, thus reducing the opportunity for entrapment. A concept of refrigerant control developed by Harnish known as the Hi/Re/Li System carries this approach one step further by operating the evaporator in a completely wetted manner. The additional heat required for vaporization is provided by the hot liquid leaving the condenser by means of a heat exchanger-accumulator combination. A scheme of this type would more nearly fulfill the oil return requirements, require no additional piping, and, according to Harnish, possesses additional benefits as well. While the operating characteristics of the Hi/Re/Li System are discussed in greater detail in the controls section, it would seem that this approach, perhaps used in conjunction with an oil separator, would be the most satisfactory from the standpoint of insuring adequate compressor lubrication.

Refrigerant Control System

The control system, in order to insure maximum effectiveness of the solar assisted heat pump, must maintain proper refrigerant flow through the system under a very large range of operating conditions. The heat pump built at North Carolina State University was originally equipped with a thermostatic expansion valve and a manual suction gas throttling

valve as the only means of refrigerant control. After determination that the thermostatic valve was malfunctioning, it was replaced with a manual expansion valve. Experience gained with this type of control system during the testing program has provided the author with a greater insight of the requirements of an automatic control system, the fundamental characteristics of which are discussed in this section.

While control of the cooling mode operation requires careful attention, heating mode operation imposes a more severe burden on the control system and therefore should be the dominant consideration in selection of controls. The primary variables encountered in heating mode operation are (1) the evaporator load as determined by solar energy availability and, to a lesser extent, (2) the condensing temperature as determined by the temperature of the storage medium.

Variations in the solar load may be divided into the following categories:

(1) Very long term due to monthly variations in solar availability.

(2) Relatively long term due to daily variations in solar availability.

and

(3) Short term due to constantly changing solar availability on partly cloudy days.

A further complication of the load situation described above arises with regard to the magnitude of the fluctuations within each category. Short term variations can often approach, in amplitude, the range of daily loads, but in a nearly stepwise manner as, for example, when dark clouds move across the sky during windy, partly cloudy conditions. Control requirements for the very long term variations are not as significant from

a design standpoint in that they are satisfied by a system fulfilling the demands encountered in categories (2) and (3).

Two basic means of control are most commonly used in refrigeration systems. They are (1) compressor capacity adjustment and (2) refrigerant flow control by means of valves within the piping system. In order to operate at as high an efficiency as possible, compressor capacity control is necessary for large load variations. Refrigerant flow control by means of adjustable expansion devices is required for smaller variations to eliminate the need for continual adjustment of the compressor capacity (which may not even be possible). Therefore, it is felt that the most satisfactory approach is to establish a balance between these two basic means of refrigerant flow control.

Compressor Capacity Control

Several alternative methods of compressor capacity reduction may be employed in the design of refrigeration systems. Since centrifugal compressors are generally applicable only in large industrial installations, this discussion is limited to positive displacement compressors. The means most frequently used for positive displacement machines are (1) variable compressor speed, (2) intermittent running (possibly with multiple compressors), (3) suction valve unloading, (4) hot gas by-pass, and (5) suction line throttling. The first three of the above mentioned methods alter capacity by changing compressor displacement. The remaining two methods accomplish their purpose by altering the characteristics of the refrigerant flow through the compressor. Haseltine and Qvale made a thermodynamic performance study of the latter three reduction techniques with additional comments regarding the others. Some

conclusions with respect to performance as well as the relative advantages of these methods follow:

1. Variable speed

   Variable speed is the most thermodynamically efficient method, but suitable drives are expensive, especially on small machines. Fixed multiple speed electric motor drives may be more readily available and economically competitive.

2. Intermittent running

   Intermittent running is regarded rough on machines if cycled too often. Stepwise reduction by the use of multiple compressors in parallel is more desirable for large capacity requirements. This would permit servicing of a compressor without total loss of system operation. Compressors may also be rotated periodically to equalize running time. Additionally, relatively large system capacities may be attained by the use of single phase machines, an important consideration for residential applications. However, high initial cost for compressors and suitable controls is a drawback. The reliability of the smaller machines when subject to heating mode operating conditions is also a consideration.

3. Suction valve unloading

   Suction valve unloading is superior to hot gas bypass and suction line throttling, but requires more complex machinery and is limited to larger installations.

4. Hot gas by-pass

   Hot gas by-pass affords little or no reduction of compressor power requirements at reduced capacity and leads to high discharge temperatures. It is superior to suction line throttling for large capacities and low pressure ratios. Advantages of this method are relative simplicity and low cost.

5. Suction line throttling

   Suction line throttling control characteristics are similar to hot gas by-pass with the exception that it does not lead to excessive discharge temperatures. It is more efficient at low capacities and high pressure ratios.

Consideration of the various characteristics stated above suggest that multiple speed or suction valve unloading would probably be the most suitable method of capacity reduction for large commercial or industrial applications where three phase power is available.

However, it is felt that intermittent running of multiple compressors is the most desirable alternative for residential applications. A schematic of a possible arrangement is shown in Figure 21. Stepwise reduction of capacity is accomplished by pressure switches which are set at varying levels and operate by sensing the suction line pressure. A reduction in the evaporator load is accompanied by a drop in the suction line pressure which eventually results in shutting off a compressor. This process continues until compressor capacity has been reduced to the required level to match the evaporator load. The stepwise capacity

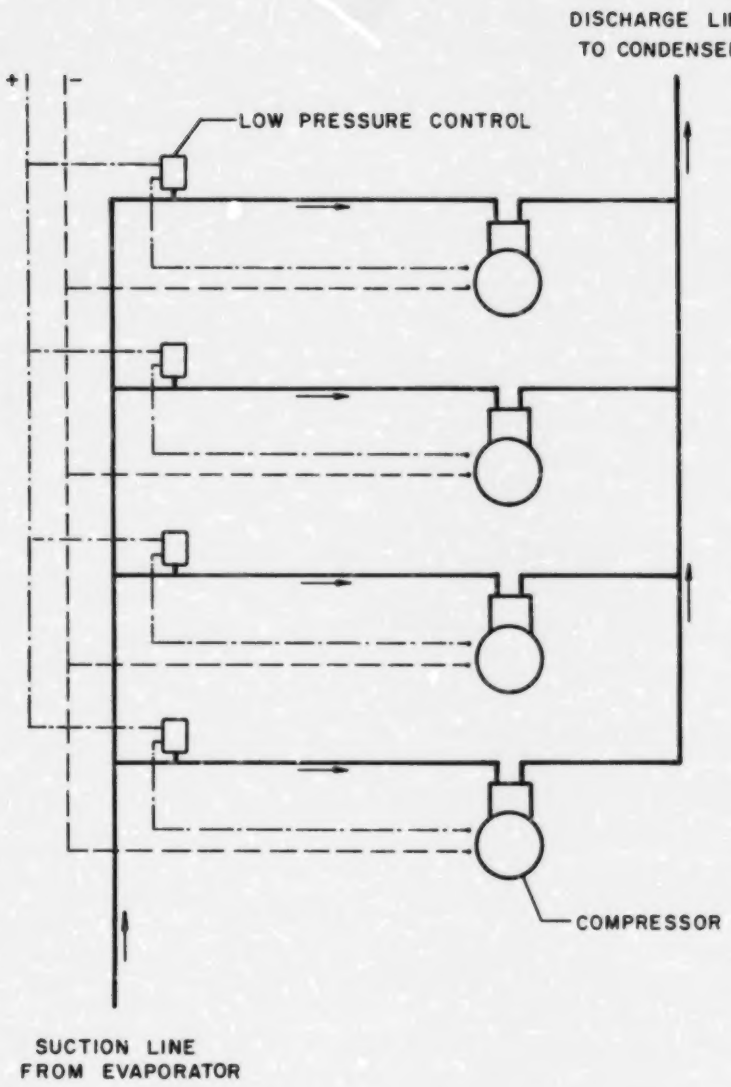


FIGURE 21. Multiple Compressor Capacity Control.

reduction perhaps may be attained more easily through the use of a staged thermostat that senses suction line temperature and activates or deactivates compressors accordingly.

Refrigerant Flow Controls

Most refrigerant flow controls in use today may be classified according to the type of evaporation technique employed in the refrigeration system, of which there are two types, the flooded system and the dry expansion system. The dry expansion system is characterized by metering to the evaporator only that amount of refrigerant needed to handle the load so that the vapor leaving the evaporator is in a somewhat superheated state. One consequence of this approach is that the liquid level in the evaporator varies with large, long term-load fluctuations, the more heat to be absorbed, the higher the liquid level. Under maximum load conditions the evaporator is nearly full of liquid, while at reduced loads the liquid level may be quite low. Operation of the dry expansion system at the reduced loads often encountered in the solar-assisted heat pump has two disadvantages. First, a large portion of the evaporator surface is not fully wetted, resulting in poor heat transfer to the refrigerant. Second, and perhaps of greater significance, is the tendency for compressor oil to become trapped in the evaporator, of particular importance in view of the high vertical rise of the solar collectors.

In dry expansion systems operating under varying loads, the amount of liquid entering the evaporator is generally controlled by means of a thermostatic expansion valve which is set to maintain suction gas superheat within narrow limits. This is accomplished by means of a refrigerant

charged bulb attached to the exit line of the evaporator which responds to changes in temperature. In the solar-assisted heat pump, however, the quickly changing solar availability may result in poor valve response. For example, if the load is suddenly reduced as when the sun becomes blocked by a cloud, time lags within the system may result in liquid leaving the evaporator before the expansion valve has had an opportunity to decrease the refrigerant flow. This situation necessitates the addition of a suction line accumulator to protect the compressor from liquid flood-back. On the other hand, by the time the valve has responded to the reduced load, that cloud may have moved on, resulting in excess superheat. Thus, under quickly varying load conditions, a thermostatic expansion valve has a tendency to "hunt", resulting in poor evaporator performance. However, system performance itself may not suffer to the same degree since the accumulator provides a thermal capacitance (see page 35). A possible means of reducing expansion valve hunting in a system of this type would be to use a valve that would directly respond to changes in solar insolation. This might be accomplished by using an electronic valve coupled with a solar cell or similar type device.

Many of the drawbacks of the dry expansion system may be eliminated through the use of a flooded evaporator (see Figure 22). In this approach, the liquid level in the evaporator is kept within narrow limits, typically by means of a low side float valve located in an accumulator or surge drum that is in parallel with the evaporator. Suction vapor is generally drawn off the top of the evaporator or accumulator. Compressor oil not carried with the vapor may be metered to the suction line

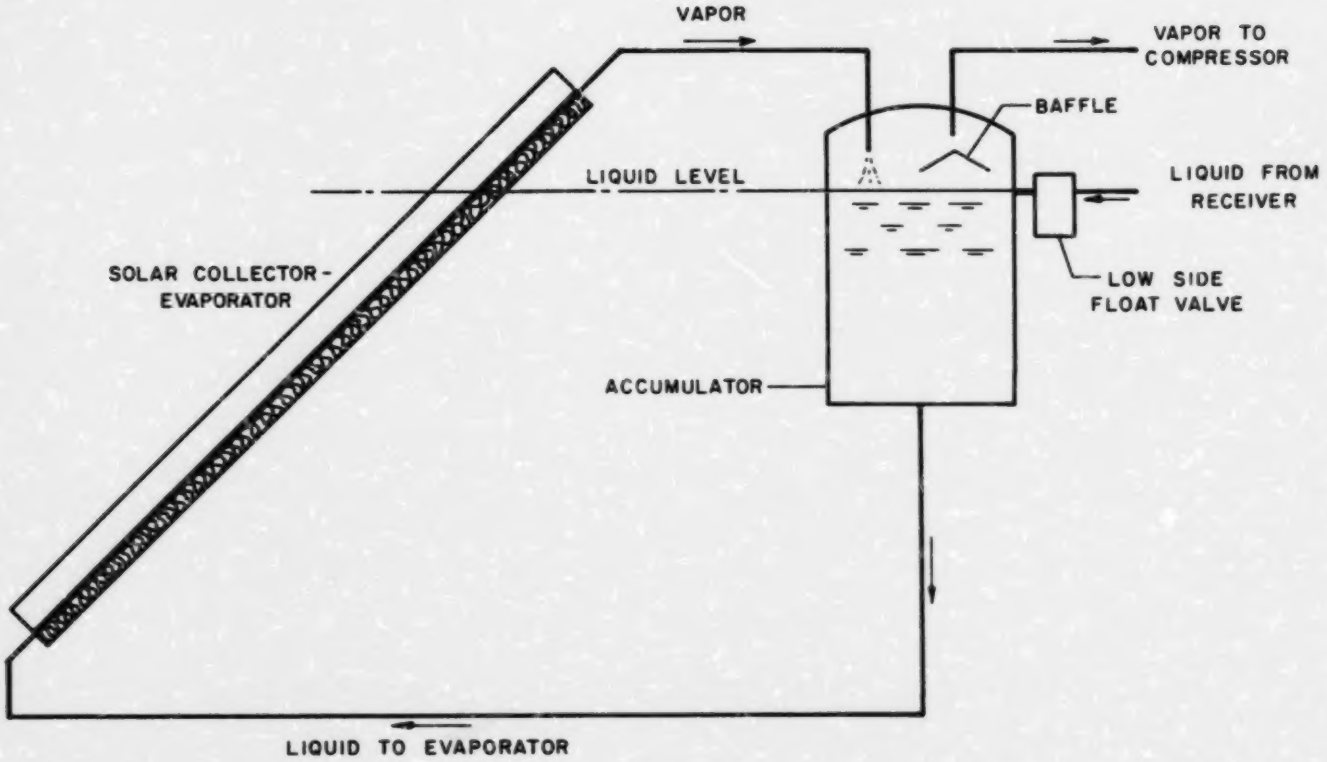


FIGURE 22. Flooded Evaporator System.

through a bleed hole or returned to the compressor through a trap, depending on the refrigerant and type of compressor in the system. Principal advantages of the flooded system include the following:

(1) more reliable oil return, as discussed in the previous section,

(2) improved heat transfer since evaporator surfaces are more completely wetted,

(3) more flexibility with regard to refrigerant distribution when several evaporators are used in parallel,

(4) little or no superheat, thus keeping compressor discharge temperatures low, especially important for suction gas cooled hermetically sealed units,

(5) somewhat lower evaporator temperatures, resulting in improved solar collector efficiency,

and

(6) a ready supply of liquid refrigerant to meet fluctuating loads (system will not hunt).

As may be noted from the above, it is felt that the flooded system is a more advantageous choice for a solar-assisted heat pump installation. Difficulties may arise, however, in the necessary vertical placement of the accumulator with respect to the solar collector-evaporators. Additionally, another accumulator would be necessary for cooling mode operation unless the condenser was placed at nearly the same height as the desired liquid level in the solar collector (during the heating mode). While these drawbacks are clearly not serious, they may be avoided by adoption of the Hi/Re/Li System mentioned in the previous section. Since the author has found little indication of widespread familiarity with this system, its operation is described somewhat in detail.

Hi/Re/Li Refrigerant Control System

Schematics of the Hi/Re/Li System's heating and cooling cycles are shown in Figures 23 and 24. The basic departures from a conventional refrigeration system found in this scheme are the use of an accumulator-heat exchanger combination and a liquid subcooling control valve. Consider the heating cycle. Hot liquid leaving the condenser, subcooled only a small amount, say 6°C, flows through the manifold check valve, on through a small heat exchanger in contact with the suction line, and finally to the accumulator-heat exchanger combination. Here the hot liquid, already slightly subcooled by the suction line vapor, gives up additional heat to the relatively cool liquid stored in the accumulator and leaves substantially subcooled. After flowing through a filter drier, the liquid is expanded through the subcooling control valve to evaporator pressure. Since the liquid entering the control valve is subcooled much more than is customary in conventional systems, each pound of refrigerant circulated through the evaporator has a greater heat absorbing capability. However, when the compressor is in balance with the condenser and evaporator (components sized correctly), it circulates only a fixed amount of refrigerant vapor for each ton of refrigeration. Thus, a greater amount of refrigerant is fed to the evaporator than can be vaporized by the heat available, resulting in a wet mixture leaving the unit. The mixture of vapor and liquid flows from the evaporator to the accumulator-heat exchanger. Here the additional heat required for vaporization is provided by the hot, high pressure liquid discharged from the condenser. Since excess refrigerant is stored in the accumulator (it functions much like a liquid receiver in this regard), only

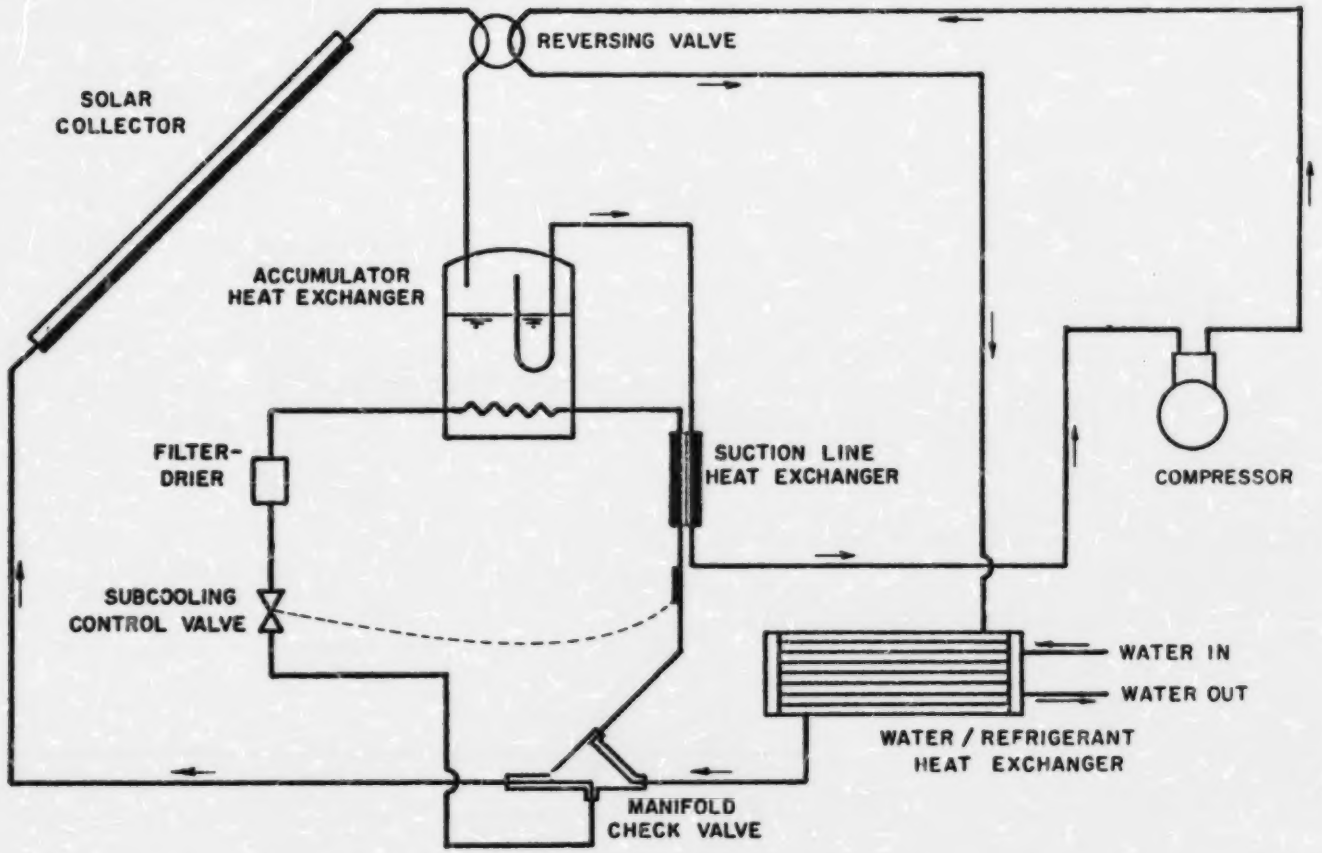


FIGURE 23. HI/Re/LI System Heating Mode Schematic.

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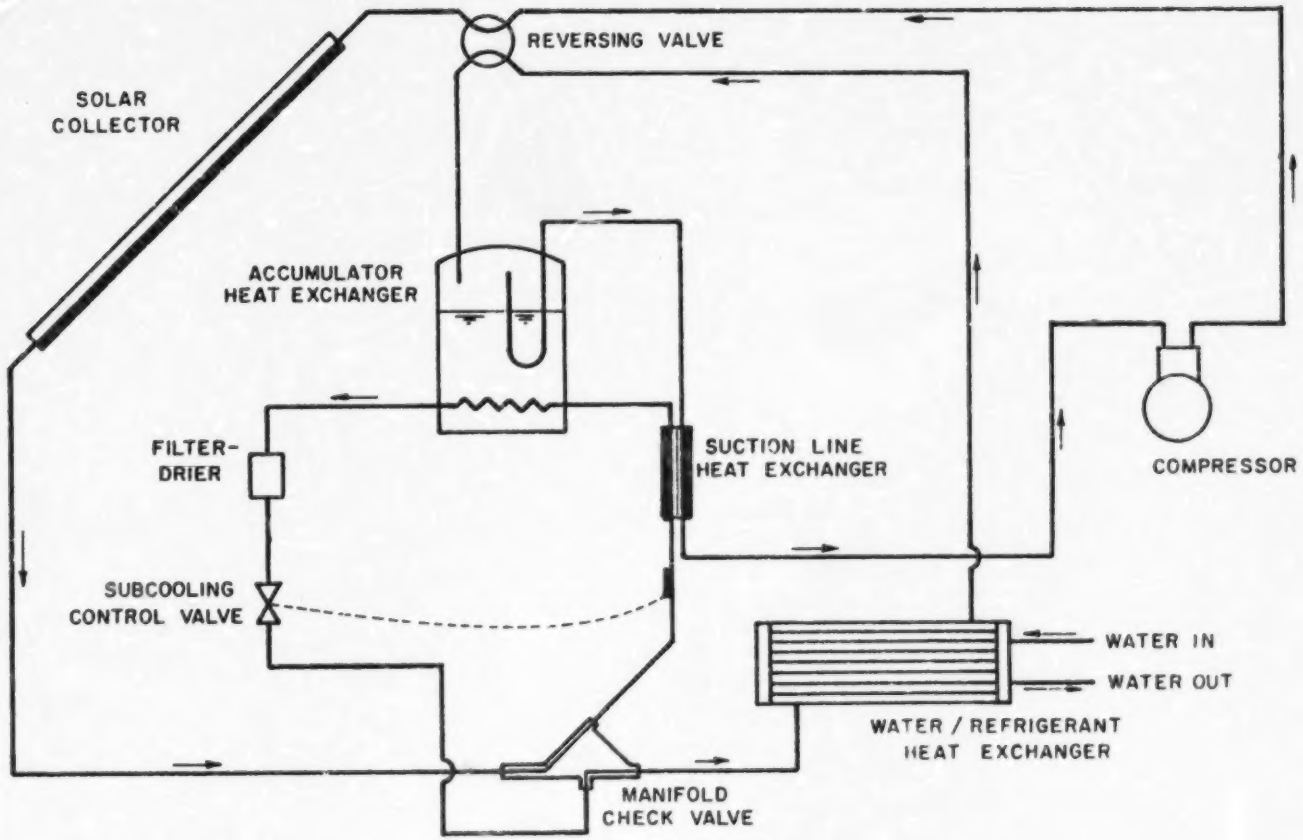


FIGURE 24. Hi/Re/Li System Cooling Mode Schematic.

slightly saturated vapor is drawn through the U-tube to the compressor. Oil and some liquid are slowly metered to the suction line through a small bleed hole at the bottom of the U-tube. The small amount of liquid refrigerant in the suction line is subsequently vaporized by heat absorbed in the small heat exchanger. Thus, there is minimal superheat of the suction gas and oil is continuously returned to the compressor.

Basic refrigerant flow control in this system is provided by the liquid subcooling control valve. It functions as an ordinary expansion valve, but rather than responding to evaporator conditions, its purpose is to maintain a prescribed degree of subcooling of liquid leaving the condenser. The valve does this by sensing condenser pressure on one side of a diaphragm and bulb pressure as determined by liquid temperature on the other. If subcooling is inadequate, the valve closes somewhat, backing liquid up into the condenser until the proper degree of subcooling is achieved. Flow balanced with the compressor pumping rate is thus acquired while maintaining optimum conditions in the condenser. Since the condenser is not normally subject to sudden changes (thanks to nearly constant storage medium temperatures during the heating cycle or slowly changing ambient temperatures during the cooling cycle), the valve has no tendency to "hunt" and smooth control is achieved.

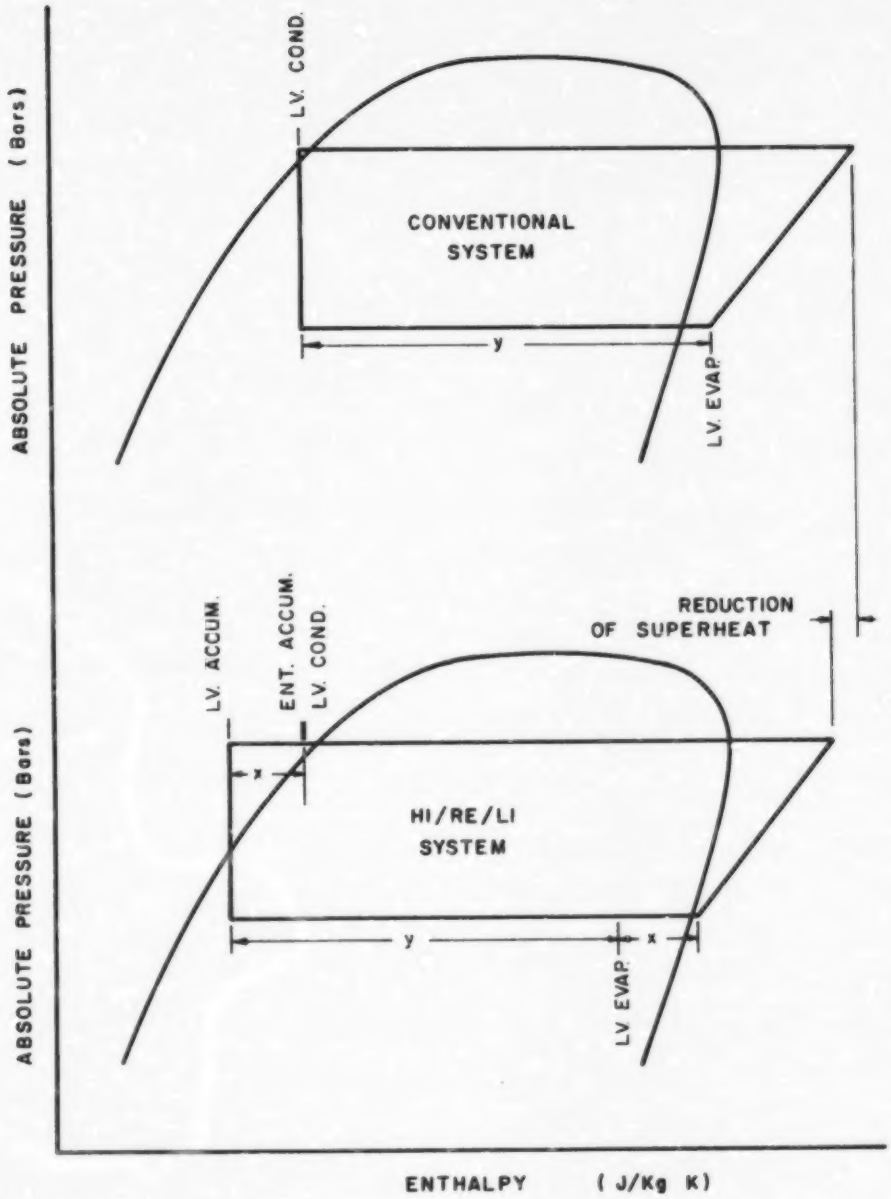
Cooling cycle operation is essentially identical to the heating cycle, the evaporator and condenser switching roles as the reversing valve is changed. The manifold check valve operates automatically, responding to the pressure reversals in the system. Any change in refrigerant requirements is accommodated by excess refrigerant stored in the accumulator.

Deviations from the conventional dry expansion refrigeration cycle associated with the Hi/Re/Li System can best be explained with the aid of the pressure-enthalpy diagrams shown in Figure 25. The evaporating temperatures and pressures are identical for both the conventional and Hi/Re/Li System. "Y" indicates the heat absorbed in the evaporator (the same for both systems) while "X" represents the heat exchanged in the suction line and accumulator heat exchangers employed in the Hi/Re/Li System. Following the condensing line on the Hi/Re/Li System diagram, one can see the effect of the additional heat exchangers as the refrigerant is further subcooled by the amount "X". Thus, substantially less flash gas is produced during expansion and an almost completely liquid mixture is fed to the evaporator. Since the evaporator can only provide "Y" amount of heat (actually somewhat more due to improved heat exchange), the refrigerant is not completely vaporized when it enters the accumulator. Here, the additional heat required for complete vaporization is provided by the hot, high pressure liquid via the heat exchanger located in the accumulator. The suction gas leaving the accumulator is essentially saturated (not superheated the customary 5-8^{o}C). If the small heat exchanger in the suction line is properly sized, the refrigerant enters the compressor only slightly superheated and compressor discharge temperatures are thereby kept as low as possible.

Advantages of the Hi/Re/Li System for application to the solar assisted heat pump are as follows:

(1) the completely wetted evaporator results in improved heat transfer and reduces the likelihood of oil entrapment,

(2) liquid is not allowed to back up in the condenser,



"x" = HEAT EXCHANGE IN SUCTION LINE AND ACCUMULATOR HEAT EXCHANGERS

"y" = HEAT ABSORBED IN EVAPORATOR

FIGURE 25. Pressure-Enthalpy Diagrams for the Hi/Re/Li and Conventional Systems.

thus optimizing its performance through more complete use of surface,

(3) oil return to the compressor is continually provided without the need for an oil separator, although one could be provided to reduce the amount of oil circulated in the system,

(4) compressor temperatures are kept as low as possible, thus extending unit life and reliability,

(5) the compressor is protected from liquid flood-back,

and

(6) reverse operation is simplified, not requiring additional controls or accumulators.

Summary of Control System Recommendations

Since numerous load fluctuations are encountered in a solar assisted heat pump, the control system must posses a high degree of flexibility, yet not be too sensitive to the very short term load variations. For residential applications, the Hi/Re/Li System of refrigerant control coupled with the use of multiple compressors is recommended. A schematic of such a system is shown in Figure 26. The use of multiple subcooling control valves is necessary to provide adjustment to large variations in system capacity. The solenoid valves are open when the respective compressor is in operation.

The short term load variations expected on partly cloudy days are effectively damped by the thermal capacitance provided by the flooded evaporator and the accumulator. Verification of the ability of an accumulator to fulfill this function is demonstrated by performance test results of the heat pump system. Figure 20 shows variations of solar insolation, upper panel temperature, and compressor inlet temperatures as a function of time for a partly cloudy day. Small variations in load

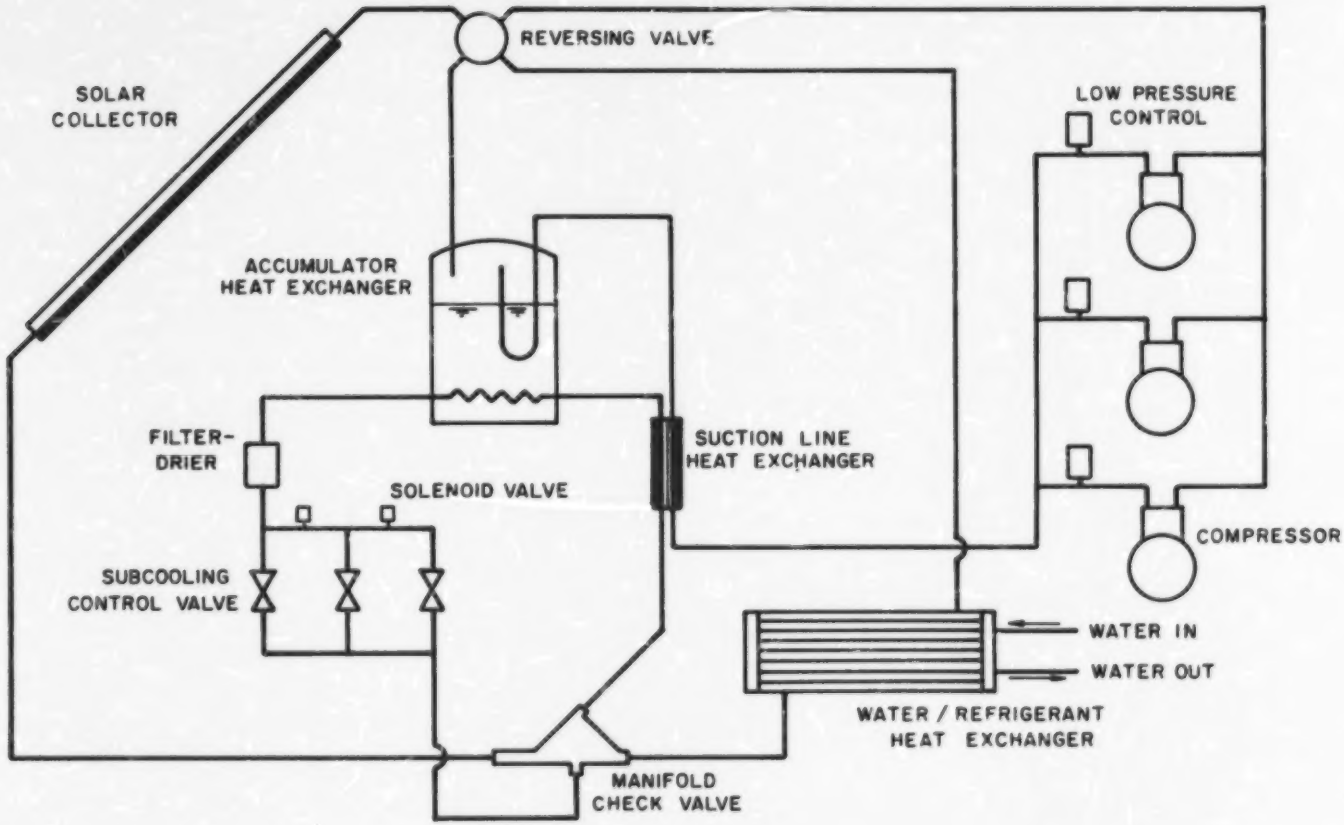


FIGURE 26. Proposed Control System Layout for a Residential Solar-Assisted Heat Pump.

are handled by the liquid subcooling valve while the larger daily variations are accommodated by altering the total compressor capacity. Thus, a balanced, smoothly operating control system that maintains the highest overall system efficiency is hopefully achieved.

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APPENDIX

HEATING MODE PERFORMANCE TEST DATA

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DATE: 7/14/76 HOUR	COMPRESSOR MOTOR POWER (watts)	HIGH SIDE PRESSURE BARS ABSOLUTE	LOW SIDE PRESSURE BARS ABSOLUTE	WATER FLOW RATE Kg/hr	WATER TEMPERATURE CONDENSER INLET °C	WATER TEMPERATURE CONDENSER OUTLET °C	REFRIGERANT TEMPERATURE CONDENSER INLET °C	TEMPERATURE CONDENSER OUTLET °C	TEMPERATURE EXPANSION VALVE °C	TEMPERATURE PANEL INLET °C	TEMPERATURE LOWER PANEL °C	TEMPERATURE MIDDLE PANEL °C	TEMPERATURE UPPER PANEL °C	TEMPERATURE PANEL OUTLET °C	TEMPERATURE COMPRESSOR INLET °C	TEMPERATURE COMPRESSOR OUTLET °C	SOLAR INSOLATION W/m^2	AMBIENT TEMPERATURE °C
1:45 PM	2722	7.98	2.12	592.5	10	22.2	89.4	39.4	39.4	16.7	18.9	20	20	19.4	46.2	180.2	607.9	60.2
2:00 PM	2550	7.98	1.91	592.5	11.1	21.7	82.8	38.3	37.8	14.4	16.7	17.8	17.8	17.8	51.2	164.2	668.7	61.2
2:20 PM	2361	7.84	1.74	592.5	10	20	82.1	36.1	36.1	11.7	12.2	12.8	12.8	12.2	44.2	166.2	547.1	61.2
2:30 PM	2361	8.87	1.91	592.5	10	19.4	86.7	36.7	34.4	12.2	12.8	13.3	43.3	31.1	63.2	179.2	688.9	61.2
2:45 PM	2361	8.74	1.70	592.5	10	19.4	91.1	36.7	35.6	11.7	12.8	13.3	25	19.4	56.2	185.2	648.4	61.2
3:00 PM	2390	8.60	1.77	592.5	9.4	20	88.9	37.2	36.7	12.8	14.4	15	17.8	13.3	52.2	180.2	638.3	62.2
3:15 PM	2351	8.46	1.70	592.5	9.4	19.4	85	36.7	35	11.7	11.7	12.2	13.9	12.2	48.2	175.2	506.6	62.2
3:30 PM	2325	8.25	1.70	592.5	9.4	18.3	82.2	34.4	32.2	8.9	8.9	8.9	12.8	12.2	40.2	168.2	567.4	62.2
3:45 PM	2024	7.91	1.56	592.5	9.4	16.7	90	31.1	29.4	5.6	6.1	6.1	38.3	33.9	74.2	188.2	466.1	62.2
4:00 PM	2400	8.60	1.91	592.5	9.4	20	85.5	37.2	35	12.2	12.2	12.2	12.8	11.7	45.2	174.2	607.9	62.2
4:15 PM	2083	7.91	1.50	592.5	8.9	17.2	81.1	33.9	32.2	7.8	7.8	7.8	8.3	7.8	36.2	167.2	466.1	62.2

DATE: 7/5/76 HOUR	COMPRESSOR MOTOR POWER (watts)	HIGH SIDE PRESSURE BARS ABSOLUTE	LOW SIDE PRESSURE BARS ABSOLUTE	WATER FLOW RATE Kg/hr	WATER TEMPERATURE CONDENSER INLET °C	WATER TEMPERATURE CONDENSER OUTLET °C	REFRIGERANT TEMPERATURE CONDENSER INLET °C	TEMPERATURE CONDENSER OUTLET °C	TEMPERATURE EXPANSION VALVE °C	TEMPERATURE PANEL INLET °C	TEMPERATURE LOWER PANEL °C	TEMPERATURE MIDDLE PANEL °C	TEMPERATURE UPPER PANEL °C	TEMPERATURE PANEL OUTLET °C	TEMPERATURE COMPRESSOR INLET °C	TEMPERATURE COMPRESSOR OUTLET °C	SOLAR INSOLATION w/m^2	AMBIENT TEMPERATURE °C
10:45 AM	2605	8.60	1.98	592.5	20.6	32.2	93.9	50.6	50.6	26.7	29.4	30.6	30.6	28.9	25.6	96.1	669.0	28.3
11:00 AM	2605	8.60	2.05	592.5	21.1	32.2	87.8	50.0	50.0	27.8	30	31.1	31.1	31.1	26.1	90.6	689.4	30.0
11:15 AM	2528	8.53	2.05	592.5	21.1	32.2	87.2	49.4	49.4	27.2	29.4	30.0	30.0	28.9	26.1	90.0	698.7	31.7
11:30 AM	2688	8.67	2.08	592.5	21.1	32.2	87.2	50.0	50.0	28.9	31.1	32.2	32.2	30	26.7	90.6	719.1	32.2
11:45 AM	2706	8.94	2.15	592.5	21.1	32.7	87.8	56.7	56.7	30.6	32.2	33.3	33.3	30.6	28.9	91.7	770.2	32.2
12:00 Noon	2784	9.29	2.25	592.5	21.1	37.2	87.8	53.9	53.9	30.6	33.3	34.4	34.4	31.1	28.9	91.1	790.6	32.2
12:15 PM	2839	9.15	2.25	592.5	21.1	35.0	89.4	52.2	52.2	30.6	33.9	34.4	34.4	32.2	29.4	92.2	790.6	32.8
12:30 PM	2772	9.15	2.25	592.5	21.1	35.0	89.4	51.7	51.7	31.1	34.4	35.0	35.0	32.2	29.4	92.2	810.2	33.3
12:45 PM	2797	9.29	2.31	592.5	21.1	37.2	90.6	53.5	53.5	31.1	34.4	35.0	35.0	32.2	30	93.5	820.4	33.3
1:00 PM	2840	9.63	2.32	592.5	20.6	35.6	92.8	53.3	53.3	32.2	35.0	35.6	35.6	33.9	31.1	96.1	830.6	33.3
1:15 PM	2840	9.36	2.32	592.5	20.0	35.6	93.3	52.8	52.8	32.2	35.0	35.6	35.6	32.2	30.6	96.1	830.6	34.4
1:30 PM	2825	9.29	2.25	592.5	20.0	35.0	91.7	52.2	52.2	31.7	33.9	35.0	35.0	32.2	29.4	95	810.2	34.4
1:45 PM	2825	9.29	2.29	592.5	20.0	35.0	91.1	52.2	52.2	31.1	33.9	34.4	34.4	32.8	29.4	93.9	810.2	34.4

DATE: 7/16/76 HOUR	COMPRESSOR MOTOR POWER (watts)	HIGH SIDE PRESSURE BARS ABSOLUTE	LOW SIDE PRESSURE BARS ABSOLUTE	WATER FLOW RATE Kg/hr	WATER TEMPERATURE CONDENSER INLET °C	WATER TEMPERATURE CONDENSER OUTLET °C	REFRIGERANT TEMPERATURE CONDENSER INLET °C	TEMPERATURE CONDENSER OUTLET °C	TEMPERATURE EXPANSION VALVE °C	TEMPERATURE PANEL INLET °C	TEMPERATURE LOWER PANEL °C	TEMPERATURE MIDDLE PANEL °C	TEMPERATURE UPPER PANEL °C	TEMPERATURE PANEL OUTLET °C	TEMPERATURE COMPRESSOR INLET °C	TEMPERATURE COMPRESSOR OUTLET °C	SOLAR INSOLATION W/m^2	AMBIENT TEMPERATURE °C
10:45 AM	2481	9.45	1.98	592.5	21.7	32.8	103.3	51.1	50.6	26.7	28.9	29.4	40	32.2	30.6	107.2	729.5	29.4
11:00 AM	2458	9.36	1.77	592.5	20.6	31.7	96.7	47.8	47.8	23.3	24.4	25	26.1	22.8	28.9	100	567.4	30
11:15 AM	2616	10.05	2.01	592.5	21.1	32.2	100.6	50	49.4	27.2	30	30.6	35.6	31.1	35.6	105	749.7	30
11:30 AM	2709	9.08	2.12	592.5	21.1	33.3	95.6	57.2	52.2	28.9	32.2	32.8	33.3	32.2	32.2	98.9	790.3	31.1
11:45 AM	2666	8.80	1.98	592.5	21.1	32.8	94.4	51.1	51.1	27.8	30.6	30.6	31.1	28.9	26.7	97.8	810.5	31.1
12:00 Noon	2753	9.36	2.19	592.5	21.1	33.3	95.6	52.2	52.2	30	32.8	33.3	34.4	30.6	32.8	98.9	810.5	31.1
12:15 PM	2787	9.56	2.25	592.5	21.1	33.3	96.1	52.8	52.8	30	33.3	33.9	34.4	31.1	33.3	99.4	830.8	31.7
12:30 PM	2772	9.63	2.25	592.5	21.1	34.4	96.1	52.8	52.8	31.1	33.9	34.4	35	32.2	34.4	99.4	830.8	31.7
12:45 PM	2766	9.63	2.25	592.5	21.1	33.9	96.1	54.4	54.4	31.1	34.4	35	35	32.2	32.2	100	830.8	32.2
1:00 PM	2766	9.63	2.25	592.5	21.1	33.9	96.1	52.8	52.8	30.6	33.9	34.4	35	34.4	34.4	98.9	830.8	32.2
1:15 PM	2766	9.63	2.29	592.5	21.1	33.9	96.1	52.8	53.3	28.9	34.4	35	35	32.2	34.4	99.4	830.8	32.2
1:30 PM	2766	8.94	2.29	592.5	21.1	34.4	93.3	51.7	51.7	31.1	34.4	35	35.6	32.2	31.7	96.1	830.8	32.2
2:30 PM	2706	8.60	2.19	592.5	21.1	33.3	89.4	49.4	49.4	30	31.1	31.1	32.2	30	26.7	90.6	770.0	32.8

DATE: 7/19/76 HOUR	COMPRESSOR MOTOR POWER (watts)	HIGH SIDE PRESSURE BARS ABSOLUTE	LOW SIDE PRESSURE BARS ABSOLUTE	WATER FLOW RATE Kg/hr	WATER TEMPERATURE CONDENSER INLET °C	WATER TEMPERATURE CONDENSER OUTLET °C	REFRIGERANT TEMPERATURE CONDENSER INLET °C	TEMPERATURE CONDENSER OUTLET °C	TEMPERATURE EXPANSION VALVE °C	TEMPERATURE PANEL INLET °C	TEMPERATURE LOWER PANEL °C	TEMPERATURE MIDDLE PANEL °C	TEMPERATURE UPPER PANEL °C	TEMPERATURE PANEL OUTLET °C	TEMPERATURE COMPRESSOR INLET °C	TEMPERATURE COMPRESSOR OUTLET °C	SOLAR INSOLATION w/m^2	AMBIENT TEMPERATURE °C
10:15 AM	2529	8.67	1.63	592.5	20.6	30	96.7	44.4	44.4	20	20.6	22.2	24.4	24.4	25	101.7	587.6	25
10:45 AM	2463	9.22	1.70	592.5	20.6	30.6	97.2	45	44.4	22.2	23.9	23.9	33.3	27.8	30	102.2	658.6	25.6
11:00 AM	2463	9.49	1.77	592.5	20.6	30.6	101.1	46.1	46.1	23.9	24.4	24.4	43.3	33.3	35.6	106.7	688.9	26.1
11:30 AM	2604	9.08	1.94	592.5	20.6	31.1	94.4	49.4	48.3	26.1	28.3	30	29.4	26.7	30	98.3	770.0	26.7
12:00 Noon	2709	8.74	2.05	592.5	20.6	32.2	90.6	50.6	50	25.6	26.7	27.8	29.4	29.4	29.4	95	810.5	26.7
12:30 PM	2785	9.29	2.19	592.5	20.6	33.9	93.3	51.7	51.7	27.2	28.3	28.3	31.1	31.7	32.2	97.8	850.8	27.8
1:00 PM	2775	9.36	2.19	592.5	20.6	33.3	94.4	52.2	52.2	30	31.7	32.2	32.8	31.7	33.3	98.9	851.1	28.3
1:30 PM	2784	8.60	2.19	592.5	20.6	34.4	82.2	50.6	50	30	32.2	32.8	32.8	31.7	32.2	92.2	840.9	28.9
2:00 PM	2784	8.25	2.19	592.5	20.6	34.4	83.3	49.4	48.9	30.6	32.2	32.8	32.8	31.1	27.8	87.2	830.8	28.9

DATE: 7-20-76 HOUR	COMPRESSOR MOTOR POWER (watts)	HIGH SIDE PRESSURE BARS ABSOLUTE	LOW SIDE PRESSURE BARS ABSOLUTE	WATER FLOW RATE Kg/hr	WATER TEMPERATURE CONDENSER INLET °C	WATER TEMPERATURE CONDENSER OUTLET °C	REFRIGERANT TEMPERATURE CONDENSER INLET °C	TEMPERATURE CONDENSER OUTLET °C	TEMPERATURE EXPANSION VALVE °C	TEMPERATURE PANEL INLET °C	TEMPERATURE LOWER PANEL °C	TEMPERATURE MIDDLE PANEL °C	TEMPERATURE UPPER PANEL °C	TEMPERATURE PANEL OUTLET °C	TEMPERATURE COMPRESSOR INLET °C	TEMPERATURE COMPRESSOR OUTLET °C	SOLAR INSOLATION w/m²	AMBIENT TEMPERATURE °C
10:00 AM	2300	9.45	1.63	592.5	21.1	27.8	98.9	44.4	43.9	19.4	21.7	27.8	21.1	22.8	25	103.9	536.9	26.7
10:30 AM	2400	9.65	1.70	592.5	20.6	30.6	97.2	45.6	45	22.2	24.4	24.4	28.3	22.2	28.3	101.7	628.2	27.2
11:00 AM	2479	9.70	1.91	592.5	20.6	30.6	96.7	47.2	46.1	24.4	27.2	27.2	31.1	25	30.6	101.1	709.2	27.8
11:30 AM	2699	10.73	2.05	592.5	20.6	31.1	102.8	49.4	49.4	26.7	27.2	27.8	34.4	28.9	36.1	106.1	729.5	28.9
12:00 Noon	2785	9.08	2.12	592.5	20	32.8	93.3	51.1	51.1	29.4	32.2	32.2	32.8	30	30	96.7	810.5	29.4
12:30 PM	2088	6.87	1.36	592.5	20	28.3	82.2	41.7	41.7	18.3	19.4	19.4	21.1	20	17.8	91.1	790.3	29.4
1:30 PM	2760	10.05	2.19	592.5	20	32.2	96.1	51.1	51.1	29.4	31.1	31.7	33.3	30.6	33.3	100.6	810.5	30.6
2:00 PM	2756	9.63	2.25	592.5	20.6	33.3	95.6	52.2	51.7	30	32.8	32.8	33.3	32.2	32.7	98.9	861.2	30.6
2:30 PM	2287	6.67	1.57	592.5	20.6	29.4	85	45		19.4	20	20	20.6	20.6	20	87.8	830.8	30.6
5:00 PM	2665	8.80	2.05	592.5	20.6	32.2	90.6	50.6	51.1	27.8	29.4	30	30	28.9	28.9	93.3	729.5	30.6

DATE: 6-5-76 HOUR	COMPRESSOR MOTOR POWER (watts)	HIGH SIDE PRESSURE BARS ABSOLUTE	LOW SIDE PRESSURE BARS ABSOLUTE	WATER FLOW RATE Kg/hr	WATER TEMPERATURE CONDENSER INLET °C	WATER TEMPERATURE CONDENSER OUTLET °C	REFRIGERANT TEMPERATURE CONDENSER INLET °C	TEMPERATURE CONDENSER OUTLET °C	TEMPERATURE EXPANSION VALVE °C	TEMPERATURE PANEL INLET °C	TEMPERATURE LOWER PANEL °C	TEMPERATURE MIDDLE PANEL °C	TEMPERATURE UPPER PANEL °C	TEMPERATURE PANEL OUTLET °C	TEMPERATURE COMPRESSOR INLET °C	TEMPERATURE COMPRESSOR OUTLET °C	SOLAR INSOLATION W/m²	AMBIENT TEMPERATURE °C
10:45 AM	2480	7.22	1.67	592.5	20	28.3	88.3	46.7	46.7	22.8	25	25.6	25.6	22.8	20	88.9	628.2	
11:15 AM	2600	7.91	1.94	592.5	20	35.5	87.8	49.4	49.4	27.8	30.6	31.1	31.7	28.3	28.3	90.6	952.1	26.7
11:45 AM	2720	8.25	2.08	592.5	20	32.2	91.1	49.4	49.4	28.9	31.7	32.2	32.7	28.9	30.6	94.4	911.9	26.7
12:00 Noon	2728	8.32	2.15	592.5	20	32.2	91.1	48.9	48.9	28.9	32.2	32.7	33.3	30	30.6	9.44	972.6	26.7
12:30 PM	2784	8.80	2.22	592.5	20	33.3	92.2	49.4	49.4	30	33.3	33.9	33.9	31.1	32.2	96.1	1013.2	27.8
1:00 PM	2768	8.53	2.19	592.5	19.4	33.3	92.2	48.9	48.9	29.4	33.3	33.9	34.4	30.6	31.7	95.6	1013.2	27.2
1:30 PM	2800	8.74	2.25	592.5	19.4	33.9	92.8	48.9	48.9	29.4	33.3	33.3	34.4	31.1	32.2	95.6	1013.2	27.8
2:00 PM	2720	8.53	2.15	592.5	19.4	33.3	91.1	50	50	28.9	31.7	31.7		29.4	29.4	93.9	891.6	27.8
2:30 PM	2700	8.53	2.12	592.5	19.4	32.8	91.1	50.6	50.6	27.8	30	30	30.6	28.3	26.7	93.3	851.1	27.8
3:00 PM	2648	7.91	2.01	592.5	19.4	32.2	85	48.9	48.9	27.2	29.4	30	30	28.3	28.3	89.4	871.3	28.9
3:30 PM	2560	7.56	1.91	592.5	19.4	31.1	86.1	47.8	47.8	26.1	27.8	28.3	28.3	26.7	25.3	89.4	770.0	28.9
4:00 PM	2400	6.18	1.70	592.5	19.4	30	91.7	48.3	48.3	23.9	24.4	24.4	24.4	23.9	22.8	94.4	668.7	27.8
4:00 PM	2500	6.25	1.57	592.5	19.4	28.9	90.6	43.9	43.9	21.7	21.7	21.7	21.7	20.6	17.8	48.3	567.4	27.8

DATE: 2-10-76 HOUR	COMPRESSOR MOTOR POWER (watts)	HIGH SIDE PRESSURE BARS ABSOLUTE	LOW SIDE PRESSURE BARS ABSOLUTE	WATER FLOW RATE Kg/hr	WATER TEMPERATURE CONDENSER INLET °C	WATER TEMPERATURE CONDENSER OUTLET °C	REFRIGERANT TEMPERATURE CONDENSER INLET °C	TEMPERATURE CONDENSER OUTLET °C	TEMPERATURE EXPANSION VALVE °C	TEMPERATURE PANEL INLET °C	TEMPERATURE LOWER PANEL °C	TEMPERATURE MIDDLE PANEL °C	TEMPERATURE UPPER PANEL °C	TEMPERATURE PANEL OUTLET °C	TEMPERATURE COMPRESSOR INLET °C	TEMPERATURE COMPRESSOR OUTLET °C	SOLAR INSOLATION W/m²	AMBIENT TEMPERATURE °C
11:30 AM	2740	11.01	2.19	592.5	20	31.7	99.4	50	50	28.9	31.7	31.7	32.8	28.9	32.8	102.2	1013.2	25.5
12:00 Noon	2784	11.22	2.29	592.5	20	34.4	101.7	51.7	51.1	31.1	33.9	33.9	34.4	30.6	35	104.4	1033.4	26.7
12:30 PM	2760	8.39	2.19	592.5	20	34.3	87.2	48.3	47.8	30.6	33.3	33.9	34.4	30.6	28.9	90.6	1073.9	26.7
1:00 PM	2764	8.46	2.19	592.5	20	33.3	88.3	46.7	46.7	31.7	33.9	33.9	33.8	31.1	29.4	91.1	1053.7	28.9
1:30 PM	2780	8.53	2.22	592.5	20.6	33.9	90	47.2	47.2	31.7	34.4	34.4	35	31.7	29.4	92.8	1053.7	28.9

DATE: 6-12-76 HOUR	COMPRESSOR MOTOR POWER (watts)	HIGH SIDE PRESSURE BARS ABSOLUTE	LOW SIDE PRESSURE BARS ABSOLUTE	WATER FLOW RATE Kg/hr	WATER TEMPERATURE CONDENSER INLET °C	WATER TEMPERATURE CONDENSER OUTLET °C	REFRIGERANT TEMPERATURE CONDENSER INLET °C	TEMPERATURE CONDENSER OUTLET °C	TEMPERATURE EXPANSION VALVE °C	TEMPERATURE PANEL INLET °C	TEMPERATURE LOWER PANEL °C	TEMPERATURE MIDDLE PANEL °C	TEMPERATURE UPPER PANEL °C	TEMPERATURE PANEL OUTLET °C	TEMPERATURE COMPRESSOR INLET °C	TEMPERATURE COMPRESSOR OUTLET °C	SOLAR INSOLATION w/m^2	AMBIENT TEMPERATURE °C
10:00 AM	2348	10.46	1.70	592.5	20.6	29.4	101.1	45	45	21.7	23.3	23.3	26.7	21.7	27.2	104.4	518	28.3
10:30 AM	2452	8.94	1.81	592.5	20.6	30.6	93.3	46.7	46.7	23.9	25.6	26.1	27.8	23.9	27.2	97.2	629	28.9
11:00 AM	2628	8.25	1.99	592.5	20.6	32.2	85	49.4	49.4	28.3	30	30.6	30.6	28.3	25.6	88.3	703	28.9
11:30 AM	2688	8.67	2.12	592.5	20.6	32.8	88.9	51.1	50.6	28.9	31.7	31.1	32.2	28.9	27.2	92.8	796	30
12:00 Noon	2688	8.94	2.46	592.5	20.6	31.7	91.1	49.4	49.4	27.8	30.6	30.6	30.6	28.3	25.6	93.3	870	30
12:30 PM	2940	11.91	1.02	592.5	20.6	34.4	102.2	52.2	52.2	33.3	36.1	36.7	37.2	34.4	36.1	105	879	30.6
1:00 PM	2800	8.94	2.29	592.5	20.6	33.9	89.4	47.8	47.8	31.7	34.4	35	35	32.2	30.6	92.8	851	30.6
1:30 PM	2840	8.94	2.32	592.5	20.6	33.9	90.6	47.8	47.8	31.7	35	35.6	35.5	32.7	31.7	93.3	851	31.7
2:10 PM	2796	8.67	2.22	592.5	20.6	33.9	88.3	47.2	47.2	31.7	33.9	34.4	34.4	31.7	30	92.2	814	31.1
3:00 PM	2700	8.12	2.15	592.5	20.6	32.2	85.6	48.9	48.9	30.6	31.7	31.7	31.7	29.4	27.8	88.3	675	31.7

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DATE: 8/26/76 HOUR	COMPRESSOR MOTOR POWER (watts)	HIGH SIDE PRESSURE BARS ABSOLUTE	LOW SIDE PRESSURE BARS ABSOLUTE	WATER FLOW RATE Kg/hr	WATER TEMPERATURE CONDENSER INLET °C	WATER TEMPERATURE CONDENSER OUTLET °C	REFRIGERANT TEMPERATURE CONDENSER INLET °C	TEMPERATURE CONDENSER OUTLET °C	TEMPERATURE EXPANSION VALVE °C	TEMPERATURE PANEL INLET °C	TEMPERATURE LOWER PANEL °C	TEMPERATURE MIDDLE PANEL °C	TEMPERATURE UPPER PANEL °C	TEMPERATURE PANEL OUTLET °C	TEMPERATURE COMPRESSOR INLET °C	TEMPERATURE COMPRESSOR OUTLET °C	SOLAR INSOLATION w/m²	AMBIENT TEMPERATURE °C
10:00 AM	2200	5.15	1.43	755.9	32.2	38.9	80	39.4	39.4	21.7	20.6		20	18.3	21.1	80.6	526.8	25.6
10:30 AM	2328	6.12	1.56	755.9	40	46.7	87.2	44.4	45	23.3	23.3		24.4	21.7	23.9	87.8	628.2	26.1
11:00 AM	2580	6.94	2.01	755.9	45.6	53.9	73.3	51.7	52.2	29.4	31.1		30.6	28.9	24.4	73.3	668.7	27.2
11:20 AM	2640	7.43	2.08	755.9	50	57.8	73.3	54.4	54.4	28.3	32.8		32.2	30	26.1	74.4	673.6	27.2

DATE: 9/13/76 HOUR	COMPRESSOR MOTOR POWER (watts)	HIGH SIDE PRESSURE BARS ABSOLUTE	LOW SIDE PRESSURE BARS ABSOLUTE	WATER FLOW RATE Kg/hr	WATER TEMPERATURE CONDENSER INLET °C	WATER TEMPERATURE CONDENSER OUTLET °C	REFRIGERANT TEMPERATURE CONDENSER INLET °C	TEMPERATURE CONDENSER OUTLET °C	TEMPERATURE EXPANSION VALVE °C	TEMPERATURE PANEL INLET °C	TEMPERATURE LOWER PANEL °C	TEMPERATURE MIDDLE PANEL °C	TEMPERATURE UPPER PANEL °C	TEMPERATURE PANEL OUTLET °C	TEMPERATURE COMPRESSOR INLET °C	TEMPERATURE COMPRESSOR OUTLET °C	SOLAR INSOLATION W/m²	AMBIENT TEMPERATURE °C
10:00 AM	2140	4.46	1.36	755.9	23.9	32.2	73.9	33.3	33.3	22.8	18.3		18.3	18.3	13.9	75	496.4	20.6
10:15 AM	2280	4.94	1.50	755.9	27.8	35.6	70.6	37.2	36.7	25.6	23.3		22.8	21.1	21.1	71.1	615.4	21.1
10:30 AM	2480	5.63	1.81	755.9	31.7	36.1	70.6	41.7	41.1	28.9	28.9		29.4	25.6	23.9	72.8	730.9	22.8
10:45 AM	2520	5.91	1.91	755.9	36.1	40	70	45	45	30.6	30.6		29.4	27.8	22.8	70.6	683.4	22.8
11:00 AM	2580	6.32	1.97	755.9	40	48.9	68.9	50	49.4	30.6	31.1		31.1	28.3	24.4	70.0	692.9	23.3
11:15 AM	2672	6.94	2.13	755.9	44.4	52.8	69.4	54.4	53.9	31.1	31.7		32.2	30.6	26.7	70.0	711.9	23.8

DATE: 9/17/76 HOUR	COMPRESSOR MOTOR POWER (watts)	HIGH SIDE PRESSURE BARS ABSOLUTE	LOW SIDE PRESSURE BARS ABSOLUTE	WATER FLOW RATE Kg/hr	WATER TEMPERATURE CONDENSER INLET °C	WATER TEMPERATURE CONDENSER OUTLET °C	REFRIGERANT TEMPERATURE CONDENSER INLET °C	TEMPERATURE CONDENSER OUTLET °C	TEMPERATURE EXPANSION VALVE °C	TEMPERATURE PANEL INLET °C	TEMPERATURE LOWER PANEL °C	TEMPERATURE MIDDLE PANEL °C	TEMPERATURE UPPER PANEL °C	TEMPERATURE PANEL OUTLET °C	TEMPERATURE COMPRESSOR INLET °C	TEMPERATURE COMPRESSOR OUTLET °C	SOLAR INSOLATION w/m²	AMBIENT TEMPERATURE °C
1:30 PM	2712	6.18	2.08	755.9	31.1	44.4	90	46.7	44.4	30.6	34.4		34.4	31.7	27.2	89.4	869.8	26.7
2:00 PM	2840	7.36	2.37	755.9	42.8	48.9	72.2	53.9	53.9	33.8	38.3		37.8	35	30	73.3	851.1	26.7
3:00 PM	3040	9.56	2.67	755.9	60.6	69.4	80	68.3	68.3		40		39.4	37.2	33.3	81.1	871.3	26.7
3:30 PM	2800	9.70	2.60	755.9	66.1	72.8	82.2	73.3	73.3		32.8		32.8	31.7	28.9	83.9	709.2	26.7

DATE: 9/16/76 HOUR	COMPRESSOR MOTOR POWER (watts)	HIGH SIDE PRESSURE BARS ABSOLUTE	LOW SIDE PRESSURE BARS ABSOLUTE	WATER FLOW RATE Kg/hr	WATER TEMPERATURE CONDENSER INLET °C	WATER TEMPERATURE CONDENSER OUTLET °C	REFRIGERANT TEMPERATURE CONDENSER INLET °C	TEMPERATURE CONDENSER OUTLET °C	TEMPERATURE EXPANSION VALVE °C	TEMPERATURE PANEL INLET °C	TEMPERATURE LOWER PANEL °C	TEMPERATURE MIDDLE PANEL °C	TEMPERATURE UPPER PANEL °C	TEMPERATURE PANEL OUTLET °C	TEMPERATURE COMPRESSOR INLET °C	TEMPERATURE COMPRESSOR OUTLET °C	SOLAR INSOLATION W/m²	AMBIENT TEMPERATURE °C
11:35 AM	2552	5.49	2.01	755.9	26.7	35.6	76.1	38.3	38.3		32.2		31.7	28.9	24.4	77.2	777.0	23.9
12 00 Noon	2700	6.39	2.19	755.9	35.6	45.6	67.2	46.7	46.7		36.1		35	32.2	28.3	68.3	805.1	26.1
12:30 PM	2856	7.22	2.52	755.9	42.8	52.8	70	53.9	53.9		38.3		37.8	34.4	31.1	70.6	832.3	26.7
1:00 PM	2960	7.98	2.60	755.9	51.7	60.6	73.3	61.7	61.7		40.6		40	37.2	33.3	74.4	841.7	27.2

DATE: 9/22/76 HOUR	COMPRESSOR MOTOR POWER (watts)	HIGH SIDE PRESSURE BARS ABSOLUTE	LOW SIDE PRESSURE BARS ABSOLUTE	WATER FLOW RATE Kg/hr	WATER TEMPERATURE CONDENSER INLET °C	WATER TEMPERATURE CONDENSER OUTLET °C	REFRIGERANT TEMPERATURE CONDENSER INLET °C	TEMPERATURE CONDENSER OUTLET °C	TEMPERATURE EXPANSION VALVE °C	TEMPERATURE PANEL INLET °C	TEMPERATURE LOWER PANEL °C	TEMPERATURE MIDDLE PANEL °C	TEMPERATURE UPPER PANEL °C	TEMPERATURE PANEL OUTLET °C	TEMPERATURE COMPRESSOR INLET °C	TEMPERATURE COMPRESSOR OUTLET °C	SOLAR INSOLATION W/m^2	AMBIENT TEMPERATURE °C
12:45 PM	2740	5.91	2.15	755.9	26.7	35.6	79.4	43.3	43.3		36.1		35.6	32.2	28.3	81.1	917.9	23.9
1:15 PM	2820	6.05	2.39	755.9	35.6	45.6	66.7	50.6	50.6		37.8		37.2	34.4	30	68.3	892.0	23.3
1:45 PM	2968	7.91	2.53	755.9	42.8	52.8	71.7	61.1	61.1		40.6		40	36.1	32.2	72.2	880.8	24.4
2:15 PM	3120	8.94	2.70	755.9	51.7	60.6	76.7	67.7	67.7		41.7		41.1	37.8	34.4	77.2	842.9	23.9

DATE: 10-29-75 HOUR	COMPRESSOR MOTOR POWER (watts)	HIGH SIDE PRESSURE BARS ABSOLUTE	LOW SIDE PRESSURE BARS ABSOLUTE	WATER FLOW RATE Kg/hr	WATER TEMPERATURE CONDENSER INLET °C	WATER TEMPERATURE CONDENSER OUTLET °C	REFRIGERANT TEMPERATURE CONDENSER INLET °C	TEMPERATURE CONDENSER OUTLET °C	TEMPERATURE EXPANSION VALVE °C	TEMPERATURE PANEL INLET °C	TEMPERATURE LOWER PANEL °C	TEMPERATURE MIDDLE PANEL °C	TEMPERATURE UPPER PANEL °C	TEMPERATURE PANEL OUTLET °C	TEMPERATURE COMPRESSOR INLET °C	TEMPERATURE COMPRESSOR OUTLET °C	SOLAR INSOLATION W/m²	AMBIENT TEMPERATURE °C
10:30 AM	2100	3.70	1.32	755.9	14.4	22.2	67.8	21.1	21.1		19.4		41.7	30.6	25.6	90	789	10
11:00 AM	2320	4.39	1.57	755.9	21.1	30	78.3	28.9	28.9		25		25.6	22.2	20.6	79.4	867.8	11.7
11:30 AM	2480	5.08	1.79	755.9	28.9	37.8	72.2	36.1	36.1		29.4		28.3	25	22.8	72.8	922.4	13.3
12:00 Noon	2628	6.05	1.94	755.9	36.7	46.1	73.3	44.4	43.9		32.8		32.2	28.9	26.1	74.4	953.4	14.4
12:30 PM	2740	7.08	2.25	755.9	45.6	55	69.4	54.4	53.9		36.1		34.4	32.8	29.4	69.4	967.6	15.6
1:00 PM	2940	7.91	2.46	755.9	52.2	61.7	71.7	62.8	62.2		38.9		37.2	35	31.7	72.8	963.8	16.1

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DATE: 10/30/76 HOUR	COMPRESSOR MOTOR POWER (watts)	HIGH SIDE PRESSURE BARS ABSOLUTE	LOW SIDE PRESSURE BARS ABSOLUTE	WATER FLOW RATE Kg/hr	WATER TEMPERATURE CONDENSER INLET °C	WATER TEMPERATURE CONDENSER OUTLET °C	REFRIGERANT TEMPERATURE CONDENSER INLET °C	TEMPERATURE CONDENSER OUTLET °C	TEMPERATURE EXPANSION VALVE °C	TEMPERATURE PANEL INLET °C	TEMPERATURE LOWER PANEL °C	TEMPERATURE MIDDLE PANEL °C	TEMPERATURE UPPER PANEL °C	TEMPERATURE PANEL OUTLET °C	TEMPERATURE COMPRESSOR INLET °C	TEMPERATURE COMPRESSOR OUTLET °C	SOLAR INSOLATION W/m^2	AMBIENT TEMPERATURE °C
10:00 AM	1680	2.74	.74	755.9	14.4	18.3	72.8	36.1	36.1		1.67		1.1	1.1	-1.1	77.8	283.4	8.9
10:32 AM	1800	3.77	.84	778.6	18.3	22.2	74.4	24.4	24.4		3.9		2.8	2.2	0	77.2	263.8	10
11:02 AM	1820	4.05	.84	778.6	22.2	25.6	72.8	27.8	26.7		4.4		3.9	2.7	.55	72.2	283.4	11.1
11:30 AM	1920	4.46	1.01	778.6	25.6	30	70	31.1	31.1		9.4		8.9	7.2	5	71.7	354.2	12.8
12:00 Noon	1980	4.74	1.04	778.6	29.4	33.8	67.8	34.4	33.9		10.6		10	8.9	6.7	69.4	364.5	13.3
12:30 PM	1900	4.94	1.01	778.6	33.9	37.8	68.9	37.8	36.7		8.8		7.8	7.2	5	71.7	303.9	13.8
1:00 PM	1872	5.15	.91	778.6	36.1	40	71.7	40.6	39.4		8.9		8.3	7.8	3.3	74.4	233	14.4
1:30 PM	1800	5.36	.88	778.6	39.4	42.2	76.1	43.3	40.6		4.4		3.3	2.8	2.2	79.4	202.3	14.4

DATE: 11/6/76 HOUR	COMPRESSOR MOTOR POWER (watts)	HIGH SIDE PRESSURE BARS ABSOLUTE	LOW SIDE PRESSURE BARS ABSOLUTE	WATER FLOW RATE Kg/hr	WATER TEMPERATURE CONDENSER INLET °C	WATER TEMPERATURE CONDENSER OUTLET °C	REFRIGERANT TEMPERATURE CONDENSER INLET °C	TEMPERATURE CONDENSER OUTLET °C	TEMPERATURE EXPANSION VALVE °C	TEMPERATURE PANEL INLET °C	TEMPERATURE LOWER PANEL °C	TEMPERATURE MIDDLE PANEL °C	TEMPERATURE UPPER PANEL °C	TEMPERATURE PANEL OUTLET °C	TEMPERATURE COMPRESSOR INLET °C	TEMPERATURE COMPRESSOR OUTLET °C	SOLAR INSOLATION w/m²	AMBIENT TEMPERATURE °C
11:00 AM	2520	5.49	1.84	778.6	20.1	29.4	68.3	29.4	29.4		29.4		28.9	25.6	22.8	70	881	12.2
11:30 AM	2600	6.11	1.94	778.6	28.3	37.2	67.2	37.2	36.7		31.7		30.6	27.8	24.4	69.4	890	12.8
12:00 Noon	2700	7.01	2.12	778.6	36.7	45.6	70.6	44.4	44.4		33.3		32.2	30	27.2	71.7	890	13.9
12:30 PM	2740	7.56	2.22	778.6	43.3	52.2	72.2	51.1	50		35.6		34.4	31.7	28.9	73.3	871	14.4
1:00 PM	2800	8.39	2.29	778.6	52.8	60.6	75	59.4	58.3		36.1		35.6	32.8	29.4	75.6	833	14.4
1:30 PM	2800	8.87	2.32	778.6	56.7	64.4	75.6	63.9	62.2		35.6		34.4	32.2	29.4	76.7	778	14.4

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15. Supplementary Notes

Langley Technical Monitor: John Samos

16. Abstract

The performance of a heat pump, utilizing 8.16 square meters of low-cost solar collectors as the evaporator in a Freon-114 refrigeration cycle, was determined under actual insolation conditions during the summer and fall of 1976. C.O.P.'s greater than 3 were obtained with condensing temperatures around 78° C and evaporating temperatures around 27° C. Ambient temperatures were about 3° C above evaporating temperatures. Similar performance levels were obtained at other insolation and temperature conditions. Experience with the system has identified some component and system changes which should increase the obtainable C.O.P. to about 4.0. These are described along with the system's design rationale. The accumulated data are presented as an appendix.

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